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PROGRESS REPORT NO. 16

ON

STUDY OF THE VIBRATION CHARACTERISTICS OF BEARINGS

PERIOD: October 1, 1962 to November 30, 1962

Contributors

Robert J. Finkelston James L. Huang

Project Leader: R. J. Finkelston
J. L. Huang

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SKF Report AL63L008
SKF Code 61023270 - 61023253
SKF Project V-3
SKF Reg. 585 14 : 422 3

Submitted to:

U.S. DEPARTMENT OF THE NAVY
CHIEF, BUREAU OF SHIPS
WASHINGTON 25, D.C.

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SKF INDUSTRIES, INC.
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SUMMARY

1. The vibration characteristics of a NJ240 cylindrical roller bearing were studied and a comparison of these characteristics with those of the NJ256 bearings is made.

2. An improved NJ240 cylindrical roller bearing was vibration tested and the effect of waviness on the vibration level of the bearing was studied analytically in a manner similar to that of the improved spherical roller bearings.

3. The vibration of ball bearings of basic 6305 size with special design features and components of low waviness was studied and comparisons are made with bearings of various sizes having good vibrational quality.

DETAILS

1. Vibration Tests of NJ240K Cylindrical Roller Bearings

In continuation of the study of vibration generated by large cylindrical roller bearings, one bearing of 200 mm bore size was tested on the large bearing vibration tester. The test bearing, designated as NJ240-A, is of standard production quality and is the same basic type of cylindrical roller bearing as the larger NJ256 bearing. The vibration tests of the NJ240-A bearing were conducted in a manner similar to the NJ256 bearings. The same test shaft and bearing housing as those used for the 23240 spherical roller bearing tests were utilized with spacer rings to compensate for the narrower width of the NJ240. The same precautions as in the tests with the NJ256 bearings were taken to avoid misalignment.

The vibrational acceleration of the NJ240-A bearing was measured in octave bands under a 10,000 lbs radial load and at rotational speeds of 100, 300 and 800 RPM. Narrow band spectra in the 0-10 KC frequency range were recorded at 300 RPM test speed for comparison with the spectra of the NJ256 cylindrical bearing given in Progress Report No. 15.

Enclosure 1, shows a tabulation of results of the octave band vibration analysis of the NJ240-A bearing in the three measuring directions under the test conditions given in the table. Enclosures 2 through 4 are the octave band spectra in the horizontal, vertical and axial direction shown graphicall for each test speed. The speed dependence of the vibration level of the NJ240-A bearing was studied in the same manner as that of the 23240 spherical roller bearing. The speed vs. amplitude exponents were computed from the octave band readings presented in Enclosure 1 using a relationship derived from the power law, $V = K n^{\alpha}$, and given in Equation (2) of Progress Report No. 15. Enclosure 5 is a table of: the amplitude ratios of vibration levels of the NJ240-A bearing corresponding to speed ratios of 300/100 and 800/100, the speed amplitude exponents α , and the average value of the exponent α in each measuring direction and octave band.

A graph of the average exponent α vs. frequency band for the NJ240-A bearing is shown in Enclosure 6 along with the corresponding curve of the NJ256 bearings given in Enclosure 42 of Progress Report No. 15.

A comparison of the octave band vibration spectra of the NJ240-A bearing and those of a larger cylindrical roller bearing (NJ256-3) is shown on Enclosure 7 in the vertical direction at 300 RPM. The NJ256-3 is one of the production quality bearings tested which were reported in Progress Report No. 15.

Enclosure 8 shows the 0-10,000 cps narrow band spectra of the NJ240-A bearing recorded in the direction of load (vertical) and normal to load (horizontal) at 300 RPM. Enclosure 1 of Progress Report No. 15 gives the comparable spectra of a 280 mm cylindrical roller bearing.

Discussion of Results:

From Enclosures 2 through 4, the spectral distribution of vibration in the NJ240-A bearing is seen in each measuring direction at the given test speeds. The vibrational acceleration is found to increase rapidly with increasing frequency in the lower bands and then decreases in the higher frequency bands. The same Enclosures also show that the spectral shape changes as the rotational speed is varied. This is evidenced by the shift of the maximum vibration levels from the 100-800 cps range at 100 RPM into the 800-6400 cps range at 800 RPM. It appears that the NJ240-A bearing vibration spectra are influenced by the same type of speed dependent peaks which were observed in the 0-1KC narrow band spectra of the NJ256 bearing (300 RPM).

Vibration in the two radial directions exhibits generally higher levels throughout the spectrum than the axial direction.

The shapes of the spectra are nearly alike in each measuring direction for each speed condition. Considering the radial directions, the vibration amplitudes in the horizontal direction in octave bands up to 400 cps are higher than those in the vertical direction, while in the octave bands above 400 cps, the opposite is true. In most respects, the NJ240-A octave band spectra show characteristics similar to those of the NJ256 bearings.

Enclosure 5 shows the experimentally determined amplitude ratios and speed amplitude exponents. The values of the exponents α seem erratic in the octave bands from 50 to 400 cps, but the exponents for both speed ratios fit the power law function fairly well in most bands above 400 cps, i.e., the α's obtained for the 300/100 and 800/100 speed ratios are approximately the same. The reason for the deviations from the power law in the low frequency range is conceivably the same influence of the speed dependent peaks which affected the vibration level vs. speed relationship of the NJ256 bearing. (See Progress Report No. 15).

To compare the effect of speed on the vibration level of the NJ240-A bearing with the results of the NJ256 bearing speed study, Enclosure 6 may be used. A comparison of the α vs. frequency curves shows that the shapes of the curves for the two bearings are similar.

It is seen from Enclosure 7 that the spectral shapes of the NJ256-3 and the NJ240-A bearing are similar. Using this Enclosure the absolute vibration levels of the two cylindrical roller bearings may be compared at 300 RPM in the vertical measuring direction. The NJ240-A bearing has lower vibration levels in octave bands below 800 cps than the NJ256-3. In the octave bands above 800 cps, the reverse tendency is true. The corresponding comparison of the spherical roller bearings shows a similar result. Since the same test bearing housing is used for both the spherical and cylindrical roller bearings of identical size, the higher readings of the NJ240-A cylindrical bearing in the high frequency range may also be attributed to the relatively lower mass of the 200 mm bearing housing. Since only one bearing of each size was used in the comparison the significance of these findings is questionable.

The 0-10KC narrow band spectrum of the NJ240-A bearing given in Enclosure 8 shows regions of relatively high amplitude vibration in the 0-1500 cps range which may contain resonances of the bearing. The spectrum shows three peaks in this range. There is reason to believe that the two lower peaks (250 cps and 750 cps) are nonresonant, based on an analogy with the NJ256-2 bearing spectra which were found to contain speed dependent peaks of relatively high amplitude in the 0-1KC range. This will be discussed in more detail in Section 2 of this report.

The portion of the spectra lying between 1,000-10,000 cps has generally lower vibration amplitudes than the portion below 1,000 cps but shows peaks in certain areas.

The peak amplitude regions of the NJ240-A spectra and the corresponding regions of the spectra of the NJ256-2 bearing given in Enclosure 1 of Progress Report No. 15 are tabulated below:

Table of Peak Amplitude Regions

NJ240 Bearing	NJ256 Bearing
1. 1000-1500 cps	1. 650-900 cps
2. 2000-2700 cps	2. 1600-1800 cps
3. 3700-4500 cps (small peak in the horizontal direction)	
4. 5400-5800 cps (Horizontal direction only)	
5. 6000-6700 cps (Vertical direction only)	3. 5800-5900 cps
6. 7500-10,000 cps (Horizontal direction only)	4. 7000-10,000 cps

Comparing the peak regions of the NJ240 and NJ256 bearings, similarities between the two spectra can be observed although the frequencies of the peak regions are not exactly the same.

Peak regions 1 and 2 of the NJ240-A bearing spectra occur at frequencies which are approximately 1.4 times those of NJ256-2 spectra. This ratio was also observed in the case of the 23240 and 23256 spherical roller bearings.

Peaks corresponding to peaks 3 and 4 of the NJ240-A can not be observed in the 2 to 5KC region of the NJ256 bearing, possibly due to the low amplification used in recording the spectrum of the NJ256 bearing in this range. Peaks 5 and 6 of the NJ240-A spectra may correspond to peaks 3 and 4 of the larger bearing.

2. Study of Improved NJ240K Cylindrical Roller Bearings

The results obtained on improved 23240 spherical roller bearings (See Progress Report No. 15) have shown that improvements of the micro-geometry of the rolling surfaces of the spherical bearings are successful in producing spherical roller bearings with improved vibration characteristics. The same approach was used in producing a cylindrical roller bearing with better quiet running characteristics than the standard production quality bearing.

The standard production quality NJ240 bearing is manufactured with polished race grooves and rollers. The NJ240-A bearing previously tested was selected as representative of the standard quality cylindrical roller bearing and the surface waviness of its components was measured.

For NJ240 roller bearings, the orders of waviness of races and rolling elements, that theoretically generate bearing vibration at frequencies corresponding to octave band limits are given in Enclosure 9. The waviness orders were computed in the octaves between 50 and 12,800 cps for rotational speeds of 100, 300 and 800 RPM. This diagram shows that vibration levels in the 50-12,800 cps frequency range for a NJ240 roller bearing operating within the given range of rotational speeds are influenced by roller waviness in the 2-2000 wpc range, by inner ring waviness between 7-15,000 wpc and by outer ring waviness between 9-19,000 wpc.

The improved cylindrical roller bearing was assembled with specially made parts whose waviness in most waviness bands was reduced considerably. The improved bearing designated NJ240-B was manufactured with honed inner and outer races and lapped rollers.

The waviness measurements of the standard and improved NJ240 bearings are presented in Enclosure 10. The ratio between the waviness velocity levels of the improved and standard production bearings are given in Enclosure 11. These results indicate that the NJ240-B rollers show the largest improvement, with waviness reading less than 10% of those of the standard production rollers in most octave bands. The inner ring of the NJ240-B bearing reads 40-90% of the NJ240-A inner ring in octaves below 60 wpc, while above 60 wpc, it reads in the range of 15-40% of the standard quality inner ring with the exception of the two highest bands. The NJ240-B inner ring reads 1.7 and 2.7 times higher in these two highest bands than the NJ240-A bearing. The NJ240-B outer ring reads approximately 80% in octave bands below 60 wpc except for the 6-12 wpc band which shows higher waviness than the NJ240-A outer. Above 60 wpc, the NJ240-B outer ring reads generally less than 30% of the production quality outer ring. In the 1920-3840 wpc band the improved outer reads 2.2 times higher than the NJ240-A outer.

The assembled NJ240-B bearing was vibration tested under 10,000 lbs radial load at speeds of 100, 300 and 800 RPM in the same manner as the NJ240-A tests described in Section 1 of this report. The vibrational acceleration readings of these bearings obtained in the octave bands from 50-12,800 cps are given in Enclosure 12, for the three measuring directions. A comparison of the octave band spectra of the standard production and improved cylindrical bearings in the vertical direction at 300 RPM is shown in Enclosure 13. Enclosure 14 is a tabulation of the ratios between the octave band vibration levels of the NJ240-B bearing and the corresponding vibration levels of the NJ240-A bearing.

As in the case of the spherical roller bearings compared in Progress Report No. 15, the readings in the octaves below 200 cps may be significantly affected by ambient vibrations. These effects may also influence the readings in higher bands at the lowest rotational speeds (100 RPM).

In the octaves between 200 and 3200 cps the improved bearing reads with few exceptions less than 40% of the standard bearing in all three measuring directions, at rotational speeds of 300 and 800 RPM. In several of these octave bands the improved bearing reads less than 10% of the production bearing.

In the 50-200 cps range the improvement is less (possibly partly due to ambient influence). In this range the NJ240-B bearing still reads lower than the NJ240-A bearing in the two radial directions, but slightly higher than the NJ240-A bearing in the axial direction. No explanation for this higher reading can be offered at this time.

In the range above 3200 cps the vibration readings are influenced by the comparatively high race waviness readings of the NJ240-B bearing in the 480-3840 wpc range. The NJ240-B bearing reads on the average 65% of the NJ240-A bearing in this frequency range.

In order to determine the effect of the waviness of each bearing component on the bearing vibration, the analysis presented in Section 4 of Progress Report No. 15 is used. Equation (6) of that report, relating the reduction of vibration of a bearing to the reduction in waviness of each component, is repeated here as follows:

$$\left(\frac{V}{V_A}\right)^2 = Z_o^2\left(\frac{W_o}{W_{oA}}\right)^2 + Z_i^2\left(\frac{W_i}{W_{iA}}\right)^2 + Z_b^2\left(\frac{W_b}{W_{bA}}\right)^2 \qquad (1)$$

The values Z_o, Z_i and Z_b which refer to the reference bearing, in this case the NJ240 A bearing, may be calculated for the cylindrical roller bearing in question from the following equations:

$$Z_o = \left[1 + 1.674\left(\frac{W_{iA}}{W_{oA}}\right)^2 + 8.807\left(\frac{W_{bA}}{W_{oA}}\right)^2\right]^{-\frac{1}{2}} \tag{2}$$

$$Z_i = \left[1 + 0.597\left(\frac{W_{oA}}{W_{iA}}\right)^2 + 5.248\left(\frac{W_{bA}}{W_{iA}}\right)^2\right]^{-\frac{1}{2}} \tag{3}$$

$$Z_b = \left[1 + 0.178\left(\frac{W_{oA}}{W_{bA}}\right)^2 + 0.190\left(\frac{W_{iA}}{W_{bA}}\right)\right]^{-\frac{1}{2}} \tag{4}$$

The values of Z_o^2, Z_i^2 and Z_b^2 were computed for the vibration in all frequency bands measured at 300 and 800 RPM shaft speed using the measured waviness band closest to the one theoretically related to the vibration band examined. These bands are shown on Enclosure 9. The waviness bands selected are the same as those given in Enclosure 58 of Progress Report No. 15. A tabulation of the squared amplification factors Z_o^2, Z_i^2 and Z_b^2 computed using the waviness readings of Enclosure 10 is shown in Enclosure 15.

The expected contributions of changes in outer ring, inner ring and roller waviness to bearing vibration changes are given by the values of Z_o^2, Z_i^2 and Z_b^2 respectively. It is seen that the average contribution in all octaves, of outer ring waviness is 20%, of inner ring waviness 25%, and of roller waviness 55%. The contribution of roller waviness to the vibration level of the NJ240-A is found to be more substantial than either inner or outer ring waviness, which appear to have equal influence. Thus, the most effective way to improve the standard quality bearing, is to reduce roller waviness.

The NJ240-B has components with a large reduction in roller waviness along with some reduction in waviness of both rings. To determine the effect of these improvements, the theoretical vibration ratios, V_B/V_A, for the NJ240 cylindrical roller bearing were computed by evaluating Equation (1), using the Z^2 factors and waviness measurements given in Enclosures 10 and 15. These theoretical ratios are tabulated in Enclosure 16.

Enclosure 17, is a graphical comparison of the predicted and measured vibration ratios in five octave bands at 300 and 800 RPM. The experimental ratios are only given for the horizontal and vertical direction for which the theory applies. Enclosure 18 gives a tabulation of the ratios between the predicted V_B/V_A and corresponding measured V_B/V_A for further comparison.

From Enclosures 17 and 18, the theoretical vibration ratio is found to exceed the measured vibration ratio in some bands, while in other bands it is somewhat smaller than the experimental ratio. The theoretical ratio is on the average 25% higher than the measured ratio. Thus, the overall vibration reduction measured on the NJ240-B bearing is slightly greater than that predicted by theory. The shape of the octave band distribution is alike for both theoretical and experimental ratios. (As shown in Progress Report No. 15, the measured vibration reduction for the 23240 spherical roller bearing was smaller than predicted by theory.)

Considering both cylindrical and spherical roller bearings it appears that the concept of amplification factors is very useful in predicting the improvements obtained from a given reduction in parts waviness.

The amplification factors $(Z_j)^2_B$, for the NJ240-B bearing were computed according to Equation (27) of Progress Report No. 15. These factors are tabulated in Enclosure 19.

it is seen that the most important contributor to the NJ240-B vibration level is the inner ring waviness which has an average contribution of 60%. The contributions from the outer ring and roller waviness are nearly equal with average contributions of 24% and 17% respectively. Thus, to further improve the NJ240-B bearing emphasis should be placed on improvement of the inner ring waviness of this bearing.

Enclosure 20, shows the 0-10KC narrow band spectra of the NJ240-B bearing recorded in the direction of load and normal to load at 300 RPM. For comparison of the narrow band spectra of the standard and improved bearings given in Enclosures 8 and 20 respectively, a hand drawn presentation of these spectra is shown in Enclosure 21.

Upon examination of the narrow band spectra of the standard and improved bearings, it is seen that the two highest peak regions at 250 and 750 cps found in the NJ240-A spectra do not appear in the NJ240-B spectra. The absence of these peak regions may be attributed to the improvement in roller waviness of the NJ240-B bearing, by analogy with the NJ256 bearing in which roller waviness peaks were found to produce similar vibration peaks in the 0-10KC spectra. (See Progress Report No. 15, Section 1.)

On the basis of the NJ240-B spectra, the lowest resonant region of the NJ240 bearing is seen to lie in the 1,000-1,500 cps range. The 2,000-2,700 cps, 3,700-4,500 cps, 5,400-5,800 cps and 6,000-6,700 cps peak regions found in the NJ240-A bearing appear also in the spectra of this bearing. However, the spectrum in the direction normal to load for the NJ240-B bearing does not show the 7,000-10,000 cps peak area observed in the NJ240-A bearing spectra.

5. Study of Ball Bearings of Special Design with Improved Quiet Running Characteristics

The vibration level of a ball bearing is influenced not only by the micro-geometry of the rolling surfaces, but also by bearing design parameters. The most efficient approach in developing a bearing with improved quiet running characteristics is therefore to consider both of these effects.

The influence of some design parameters such as the number of balls and outer ring mass were examined analytically in the two Special Reports AL61L032 and AL62L005 and some of the findings were experimentally verified in Progress Report 9-10. The influence of micro-geometry was also studied analytically and experimentally in the reports mentioned above.

To obtain a bearing with improved vibration characteristics a new design with the following main features was developed:

1. The number of balls is increased above that used in bearings of standard design with the same boundary dimensions.
2. The thickness of the outer ring is increased. This is accomplished by reducing the ball diameter.

The new design will in this report be referred to as the "thick ring bearing*".

The "thick ring bearing" was made with the same boundary dimensions and pitch diameter as a 6305 bearing. The number of balls was increased from 7 to 15 and the ball size reduced from 7/16" to 17/64". The "thick ring bearing" was designed as an angular contact bearing and was equipped with a one piece cage made of textolite, instead of the pressed steel cage used on standard SKF 6305 bearings, to permit faster manufacturing of an experimental lot.

* It is the opinion of SKF Industries, Inc., that this design represents a patentable invention.

The increased number of balls and the thick outer ring of the "thick ring bearing" are expected, according to theory and previous experimental results, to have the following effects on the vibration characteristics of the bearing:

1. According to rigid ring theory, the vibration level generated by waviness of any element of an axially loaded bearing (in a non-resonant band) is approximately inversely proportional to the square root of the number of balls. This effect was experimentally verified and reported in Progress Report No. 9-10. The theoretical reduction in vibration level due to the increase in the number of balls from 7 (standard) to 15 (thick ring bearing) is 32%.

2. The increased thickness of the outer ring of the "thick ring bearing" makes it flexurally more rigid, and less subject to flexural vibrations due to ball loads. According to special report AL61L037 the amplitude of outer ring flexural vibrations due to ball loads is inversely proportional to the moment of inertia of the cross-section of the ring and also inversely proportional to the square of the number of balls. The vibration amplitudes of the "thick ring bearing", originating from outer ring flexural effects due to ball loads are expected to be only 9% of the amplitudes of the standard 6305 bearing. Outer ring flexural vibration due to ball load should occur mainly in the low frequency range. The fundamental frequency of the outer ring flexural vibration due to ball load is the ball passage frequency over the outer ring, which for a 6305 bearing rotating at 1800 RPM is 78 cps. Higher harmonics of the motion exist according to special report AL61L037, but their amplitudes are much lower than that of the fundamental frequency.

3. Flexural vibrations of the outer ring are induced, in addition to the vibrations due to ball loads discussed above, also by low order race waviness. The following brief analysis is given to show what vibration frequencies are expected due to these causes:

Enclosure 22 illustrates a ball bearing with geometrically perfect outer race and "wavy" inner race. The outer race is stationary and the inner ring rotates with a constant angular velocity ω_r. The outer ring deforms flexurally under the influence of the ball loads. Due to the waviness of the inner race the radial displacements of the outer ring at the different ball contacts are not the same, but depend on the relative position of the inner ring with respect to the ball set. In this analysis the Hertzian deformations at the ball contacts are neglected and the outer ring is assumed to deform only flexurally, ie., the analysis presumes a fixed outer ring center (no rigid body motion). Since, under these conditions, the relationship between load and deflection may be considered linear, the outer ring deformation at a given point O (See Enclosure 22) may be expressed in terms of the outer ring displacements $X_1, X_2, \ldots, X_j, \ldots, X_z$ at the ball locations as follows:

$$X = C_1 X_1 + C_2 X_2 + \ldots . C_j X_j + \ldots . C_z X_z \tag{1}$$

where $C_1, C_2 \ldots \ldots C_z$ are

"influence coefficients" which depend on the angular positions of balls No. 1, 2,... z, respectively, with respect to point O, but not on the values of $X_1, X_2 \ldots X_j \ldots X_z$.

The influence coefficients vary as the ball set rotates with a period corresponding to one full cage revolution. The influence coefficient C_j for the j^{th} ball may therefore be expressed by the Fourier Series

$$C_j = \sum_{m=1}^{\infty} \bar{C}_m \sin m \left[\omega_c t + \phi_n + \frac{2\pi(j-1)}{Z} \right] \tag{2}$$

where ω_c is the angular frequency of the rotating ball set and $\bar{C}_m$ and ϕ_m are constants depending on the physical characteristics of the ring.

Since the variation in Hertzian deformation at the ball contacts is neglected, the outer ring displacement at any given ball is exactly the same as the inner ring waviness displacement under the same ball. The displacement at the j^{th} ball may according to special report AL61L032 be expressed as:

$$X_j = \sum_{K=1}^{\infty} A_K \sin K \left[\omega_i t + \alpha_K - \frac{2\pi(j-1)}{Z} \right] \tag{3}$$

where ω_i is the angular frequency of the rotating ball set with respect to inner ring, and A_K is the amplitude and α_K the phase angle for the K^{th} waviness harmonic. The displacement X_j has a period corresponding to one inner ring revolution with respect to the ball set.

Inserting (2) and (3) in (1)

$$X = \sum_{K=1}^{\infty} A_K \sin K \left[\omega_i t + \alpha_K - \frac{2\pi(j-1)}{Z} \right] \sum_{m=1}^{\infty} \bar{C}_m \sin m \left[\omega_c t + \phi_m + \frac{2\pi(j-1)}{Z} \right] \tag{4}$$

Using the trigonometric identity

$$\sin\alpha \sin\beta = \tfrac{1}{2}[\cos(\alpha-\beta) - \cos(\alpha+\beta)]$$

and performing the multiplication of the summations in Equation (4), it is seen that the spectrum of X contains the frequencies

$$\omega = K\omega_i \pm n\omega_c \qquad (5)$$

Inner ring waviness of the order K (=1, 2, 3 $\cdots$) generates several pairs of frequencies centered around the frequency $K\omega_i$ and spaced $n\omega_c$ from the center point.

Since

$$\omega_i + \omega_c = \omega_r \qquad (6)$$

peaks at multiples of the rotational frequency ω_r occur in the spectrum (for $K = n$ using the positive sign in (6)).

The analysis does not give the amplitudes of the various peaks, only the expected frequencies. A more detailed analysis would possibly show that some of the peaks are much more predominant than others; some peaks may be completely missing.

Since the outer ring is comparatively stiffer in the higher flexural modes, only the lower vibration peaks are expected to be of significant magnitude, i.e., peaks generated by low order inner ring waviness, such as 2 and 3 wpc.

(The vibrations due to eccentricity, $K = 1$, are equivalent to a rigid body motion of the outer ring.)

The frequencies of the flexural outer ring vibrations induced by outer ring waviness can be obtained by replacing ω_i in Equation (4) by ω_c. The corresponding frequencies are

$$\omega = K\omega_c \pm n\omega_c = m\omega_c \qquad (m = 1, 2, 3, \cdots)$$

i.e., the spectrum generated by outer ring waviness is characterized by peaks at all multiples of the cage frequency ω_c. As in the case of inner ring waviness some of the peaks may be more predominant than others.

For experimental verification of the analytically determined peak frequencies narrow band spectra of a few 6205 and 6207 bearings with known two point and three point out-of-roundness of the inner ring, (measured using a Talyrond instrument made by Taylor, Taylor and Hobson) are shown in Enclosures 23, 24 and 25. Enclosure 23, shows the spectrum of a 6205 bearing with predominant inner ring two point out-of-roundness of 400 microinches. The three point out-of-roundness of the inner ring for this bearing is approximately 10 microinches. Since the rotational speed of the bearing was 1800 RPM, the first peak on the spectrum corresponds to the rotational frequency of the inner ring, caused by inner ring eccentricity. The second peak appearing at twice the rotational frequency, is assumed to have been induced by inner ring two point out-of-roundness, and the third, at three times the rotational frequency, by inner ring three point out-of-roundness. The fourth peak corresponds to the ball passage frequency over the outer ring, which may be induced by various causes, such as flexural vibrations of the outer ring due to the finite ball spacing or by outer ring two point out-of-roundness, or by 8 and 10 wpc outer ring waviness according to rigid ring theory. It should be noted that the ordina scale on Enclosure 23 is decibels, and the peak at twice the rotational frequency is more than 20 db (ten times) higher than t peak at three times the rotational frequency.

The spectrum of Enclosure 23 seems to indicate that the peaks at multiples of the rotational frequency are clearly predominant, i.e., 2 wpc inner ring waviness produces a vibration peak at twice the rotational frequency, 3 wpc at three times the rotational frequency. Other frequencies expected from the theory cannot be identified. The brief theory presented here does not suffice to explain why the other frequencies do not appear in the experiment. This is further illustrated by Enclosure 24, which shows an example of a spectrum where peaks at five multiples of the rotational frequency can be seen. Both the two point and three point out-of-roundness of the inner ring are in this case approximately 30 microinches. The bearing represented by the spectrum of Enclosure 25 has comparatively low two point out-of-roundness (approximately 10 microinches) and three point out-of-roundness (approximately 20 microinches). The peak at twice the rotational frequency does not appear in this case.

The thick ring design, because of the increased rigidity of outer ring, is expected to reduce the amplitudes of the flexural vibrations induced by low order waviness.

4. The resonant frequencies of the bearing outer ring are influenced by the outer ring mass and cross-section, by the spring constant of the balls contacting the races and by the number of balls. The resonant characteristics of the "thick ring bearing" are therefore, entirely different from those of a standard bearing. Since the resonances in general occur at comparatively high frequencies, the main effect of the change is resonant characteristics is expected in the high frequency range.

A sample consisting of five "thick ring bearing" was evaluated by measuring the vibration under purely axial load in the three frequency bands 50-300, 300-1800 and 1800-10,000 cps (the same as the Anderometer bands). A Bench Center Type Vibration Tester developed by SKF Industries, was used[1]. In manufacturing the thick ring bearings, efforts were made to produce a bearing with extremely low vibration level, not only by changing the design, but also by improving the micro-geometry of the parts.

(1) For details of the tester and loading method: See Final Report on Calibration of Anderometer, Submitted to U.S. Department of the Navy under Contract NObs-78593.

For comparison of the vibration levels, four samples of bearings in the 6203-6207 size range of good quiet running quality (of SKF's and competitive production) were also vibration tested.

For further evaluation of the "thick ring bearing" and the other samples used for comparison, the waviness of the races and balls and two point out-of-roundness of the race grooves was measured. The average readings of these parameters along with the average vibration readings are tabulated in Enclosure 26. The number and size of balls in each bearing is also shown. As an indicator of the flexural rigidity of the outer ring the quantity I/R^3 is listed (I is the moment of inertia of the outer ring cross-section and R the mean ring radius). The reciprocal quantity R^3/I is proportional to the ring deflection under given concentrated loads. The stiffness of the ring increases with increasing I/R^3.

Enclosure 26, shows that in the low band the average vibration level of the "thick ring bearing" is only 55% of that of the best sample of standard bearings. In the medium band the "thick ring bearing" reads approximately the same as the best standard production bearing. In the high band the "thick ring bearing" reads more than three times as high as the standard samples in the 6203 size, more than twice as high as the 6205 samples, but 20% lower than the 6207 sample.

Of the micro-geometrical parameters listed on Enclosure 26, the vibration level in the low band is influenced by race waviness in the 3-6, 6-12 and 12-24 wpc bands, by ball waviness of the order 4 wpc and by ball and race two point out-of-roundness. A comparison between sample No. 1 ("thick ring bearing") and sample No. 3 (the best production sample) shows that inner ring waviness in the three lowest bands of the two samples is approximately the same. For outer ring waviness the thick ring sample reads higher than sample No. 3 in the 3-6 wpc band, but lower in the 6-12 and 12-24 wpc bands. The ball waviness of sample No. 3 is somewhat higher than for the "thick ring bearing" in the 4-8 wpc band, and the ball two point out-of-roundness of both samples is approximately 2 microinches.

inner ring out-of-roundnesses are comparable and outer ring out-of-roundness is somewhat higher for the "thick ring bearings". Although the overall effect of waviness may slightly favor the thick ring bearing, the substantial difference between the low band vibration levels of the two samples is not explainable by the micro-geometrical parameters only. According to theory, the larger number of balls in the "thick ring bearing" may account for a reduction in the vibration level of 30%. A substantial reduction is also believed to be attributed to the increased flexural rigidity of the "thick ring bearing" outer ring as indicated by the quantity I/R^3 shown on Enclosure 26. It is seen that I/R^3 for "thick ring bearing" is four times that for sample No. 3.

A comparison between sample No. 2 and sample No. 3 shows that their low band vibration levels are essentially the same, but the race waviness and out-of-roundness values are considerably higher for sample No. 2. Both samples are 6203 bearings, but sample No. 2 has one more ball than sample No. 3 and because of its smaller ball size the outer ring of sample No. 2 is more rigid than sample No. 3, as indicated by the difference in the I/R^3 values of the two samples. This may account for the relatively low vibration level of sample No. 2 in the low band. In the medium band both samples, read approximately the same which is in agreement with the waviness measurements. In the high band sample No. 2 reads 50% higher than sample No. 3 which may be caused by the difference in the resonant characteristics of the outer ring of the two bearings. Since the same effect was observed for the "thick ring bearing", it would appear that in general an increase in the thickness of the outer ring tends to increase the vibration level in the high band.

Enclosures 27 and 28 show the vibration spectrum of the "thick ring bearing" in the 0-10,000 cps range. It is seen from Enclosure 27, that peaks appear at the rotational and twice the rotational frequency, but not at higher multiples of the rotational frequency or at the ball passage frequency.

The ordinate scale in Enclosure 27, is different from that of Enclosures 23 through 25, so that no direct comparison between the amplitudes of the various spectra can be made. The high frequency spectrum of the "thick ring bearing", on Enclosure 28 shows three high peak regions, in the 300-500 cps range, in the 3000-4500 cps range and in the 5500-6500 cps range. There are no high peaks in the 500-3000 cps range. A comparison with the spectrum of a standard 6305 bearing, given in Enclosure 26 of Progress Report No. 8, indicates that the high frequencies are more predominant in the "thick ring bearing".

The standard 6305 bearing has comparatively high amplitudes below 3000 cps, lower in the 4000-6000 cps range, and a peak region around 6700 cps, which is not quite as predominant as the 6000 cps peak region in the "thick ring bearing" spectrum.

It appears from the test results that the "thick ring" design, effectively reduces the vibration in the low frequency range, but tends to increase the high frequency amplitudes. Since vibrations in the high band are generally easier to control by micro-geometry or in assembly (See Progress Report No. 6 for housing effects) than in the low band. This disadvantage may easily be outweighed by the improvements obtainable on the low band.

Plans for the Near Future

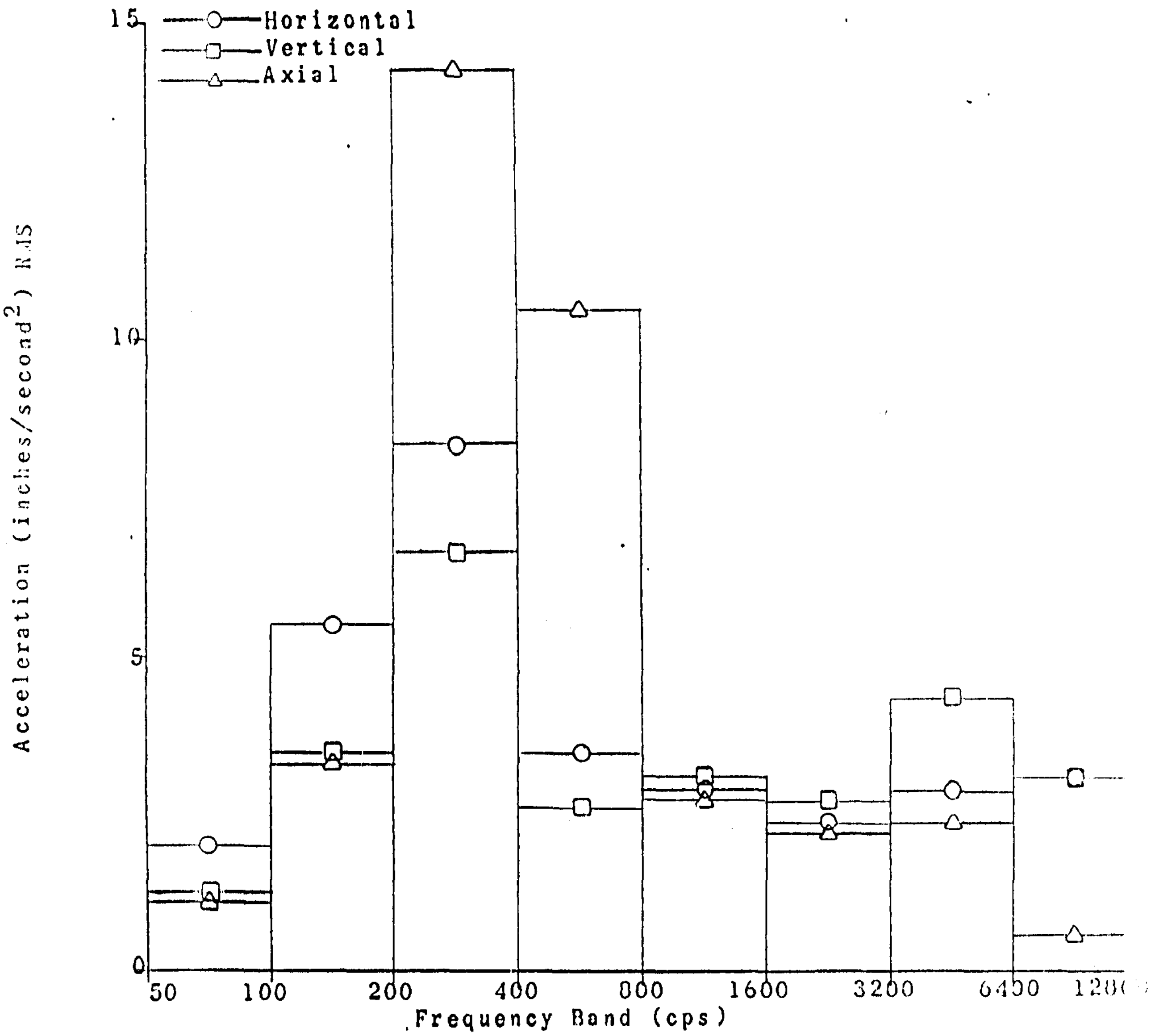
The vibration of large tapered roller bearings of different quality levels will be studied and the effects of improvement in the manufacuting of the components will be discussed.

The airborne noise of the standard quality spherical, cylindrical and tapered roller bearings will be studied and compared.

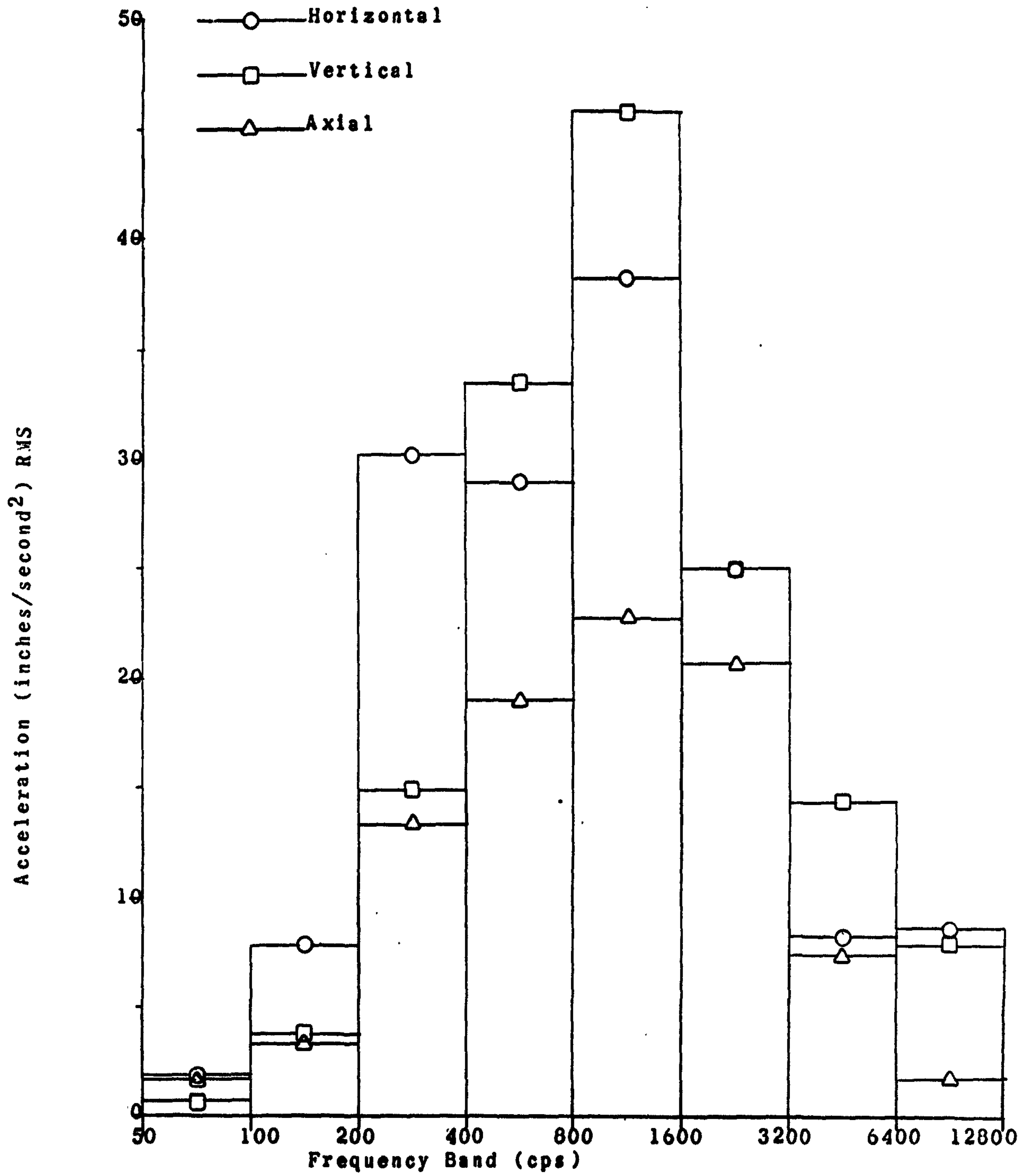
ENCLOSURE 1

VIBRATIONAL ACCELERATION MEASUREMENTS OF THE NJ240-A CYLINDRICAL ROLLER BEARING AT 10,000 LBS. RADIAL LOAD

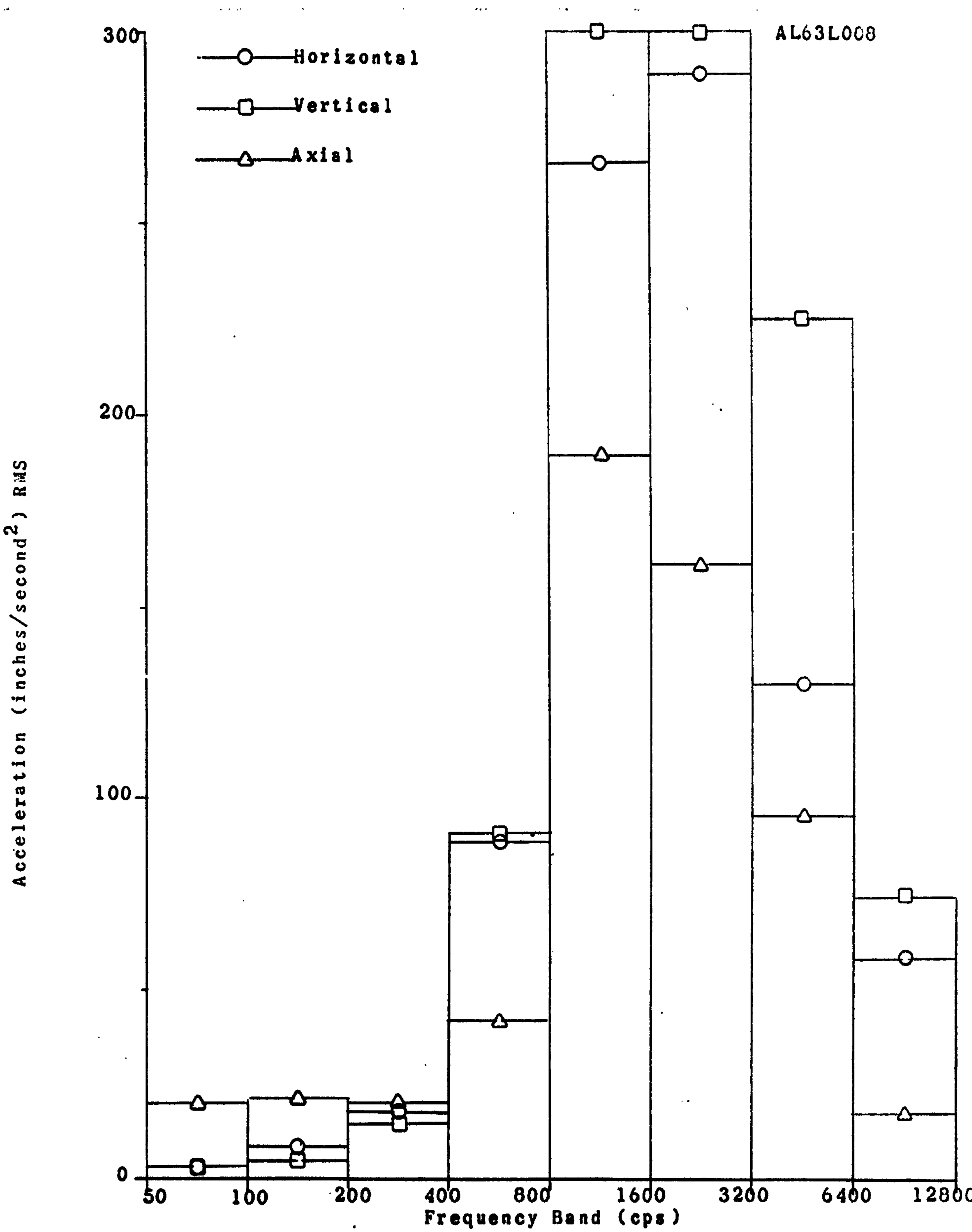
	Acceleration (inches/second^2) RMS							
Rotational Speed	50-100	100-200	200-400	400-800	800-1600	1600-3200	3200-6400	6400-12800
100 RPM								
Horizontal	1.97	5.56	8.34	3.48	2.90	2.40	2.88	3.04
Vertical	1.24	3.97	6.69	2.61	3.10	2.72	4.32	3.04
Axial	1.09	3.32	14.30	10.50	2.85	2.24	2.40	0.58
300 RPM								
Horizontal	1.74	7.88	30.20	29.00	38.30	22.40	8.31	8.63
Vertical	0.82	3.72	14.90	33.50	45.80	22.40	14.40	7.83
Axial	1.71	3.42	13.30	19.00	22.80	20.80	7.35	1.76
800 RPM								
Horizontal	3.02	8.57	19.70	88.10	266.0	288.0	128.0	57.60
Vertical	3.10	3.96	14.30	91.60	273.0	272.0	224.0	76.80
Axial	19.90	21.80	19.00	42.70	190.0	154.0	96.0	16.80



ENCLOSURE 2 OCTAVE BAND VIBRATION SPECTRUM OF THE NJ240-A CYLINDRICAL ROLLER BEARING UNDER 10,000 LBS. RADIAL LOAD AND AT 100 RPM.



ENCLOSURE 3 OCTAVE BAND VIBRATION SPECTRUM OF THE NJ240-A CYLINDRICAL ROLLER BEARING UNDER 10,000 LBS. RADIAL LOAD AND AT 300 RPM.

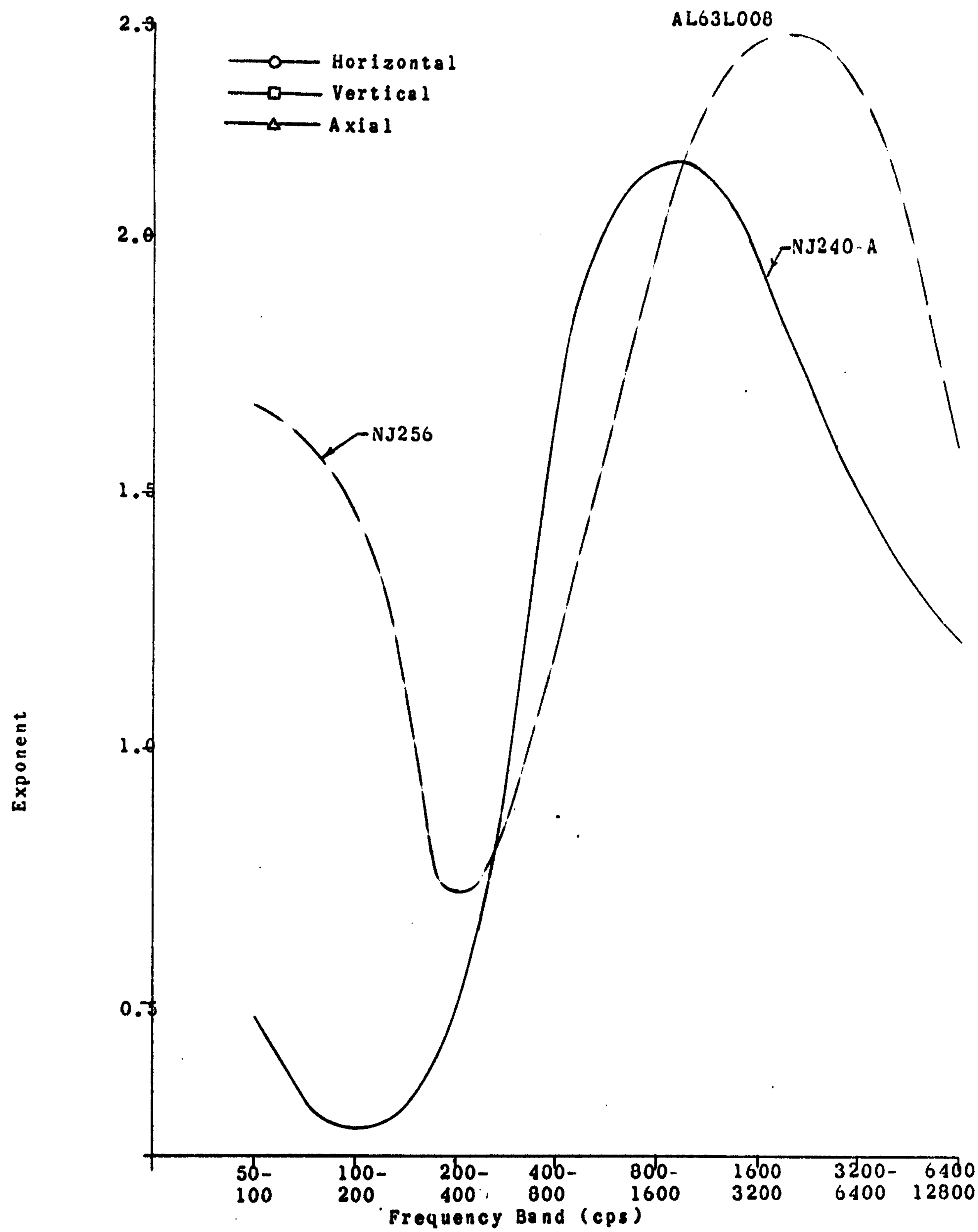


ENCLOSURE 4 OCTAVE BAND VIBRATION SPECTRUM OF THE NJ240-A CYLINDRICAL ROLLER BEARING UNDER 10,000 LBS. RADIAL LOAD AND AT 800 RPM.

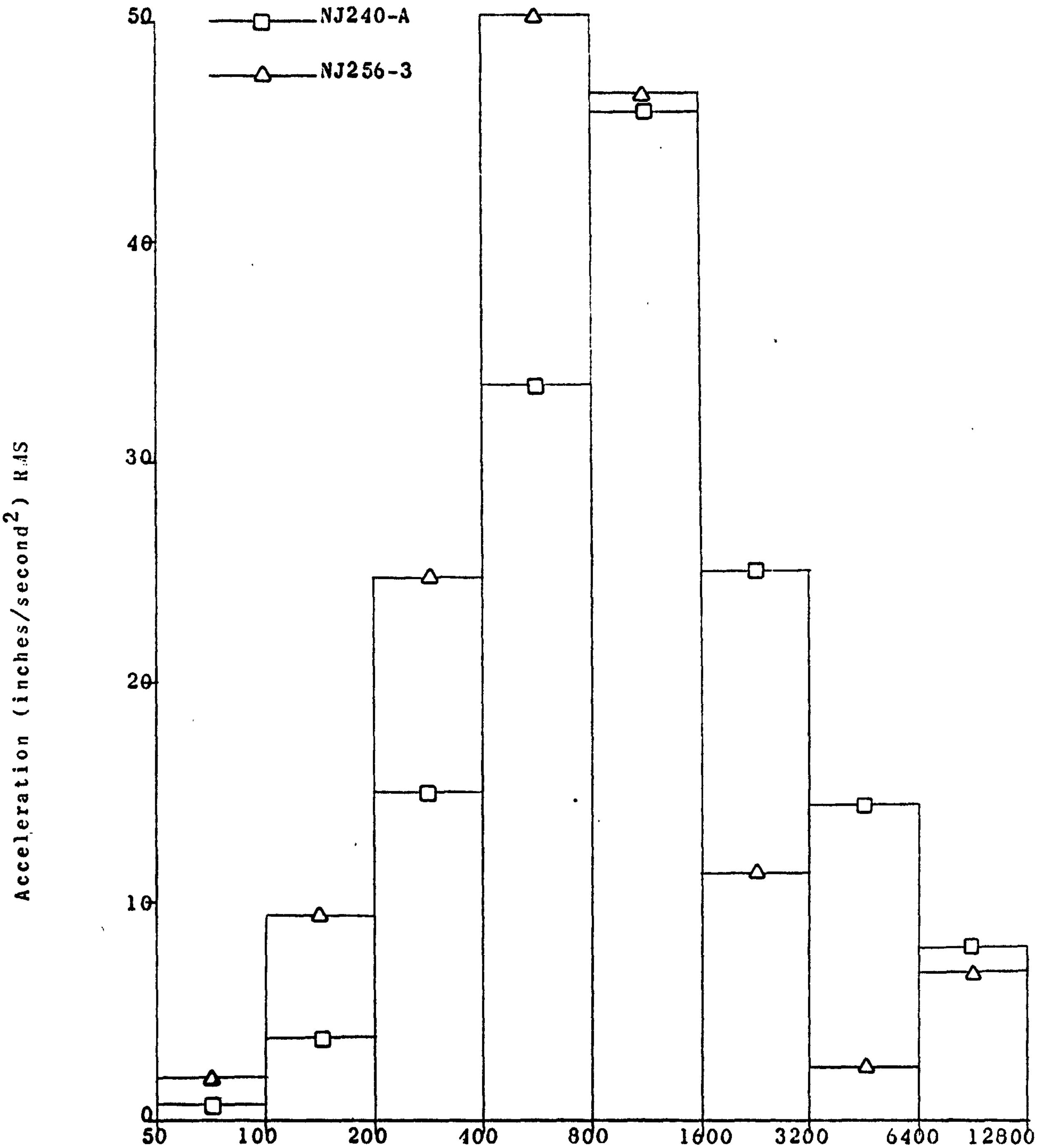
ENCLOSURE 5

TABLE OF VIBRATION AMPLITUDES RATIOS AND COMPUTED SPEED AMPLITUDE EXPONENTS
FOR THE NJ240-A BEARING AT SPEED RATIOS OF (300/100) AND (800/100)

Speed Ratios	Amplitude Ratios and Exponents	Frequency Bands (cps) 50-100	100-200	200-400	400-800	800-1600	1600-3200	3200-6400	6400-12800
	Horizontal								
	Amplitude Ratio	0.88	1.42	3.62	8.33	13.21	9.33	2.89	2.84
		-0.17	0.32	1.17	1.92	2.35	2.04	0.97	0.95
300/100	Vertical								
	Amplitude Ratio	0.66	0.94	2.23	12.84	14.77	8.24	3.33	2.59
		-0.38	-0.06	0.73	2.32	2.44	1.66	1.09	0.85
	Axial								
	Amplitude Ratio	1.57	1.03	0.93	1.81	8.00	9.28	3.06	3.06
		0.41	0.03	-0.07	0.54	1.89	2.02	1.02	1.02
	Horizontal								
	Amplitude Ratio	1.53	1.54	2.36	25.32	91.72	120.00	44.44	18.95
		0.21	0.21	0.41	1.55	2.16	2.30	1.82	1.41
800/100	Vertical								
	Amplitude Ratio	2.50	0.99	2.14	35.09	88.06	100.00	51.85	25.26
		0.44	-0.01	0.37	1.71	2.16	2.21	1.89	1.56
	Axial								
	Amplitude Ratio	18.26	6.57	1.33	4.07	66.66	68.75	40.00	29.16
		1.39	0.91	0.14	0.68	2.02	2.03	1.77	1.62
	Average Exponents								
		0.19	0.27	0.79	1.74	2.26	2.17	1.40	1.28
		0.34	-0.03	0.55	2.02	2.30	1.93	1.49	1.21
		0.90	0.47	0.14	1.22	1.95	2.08	1.39	1.32

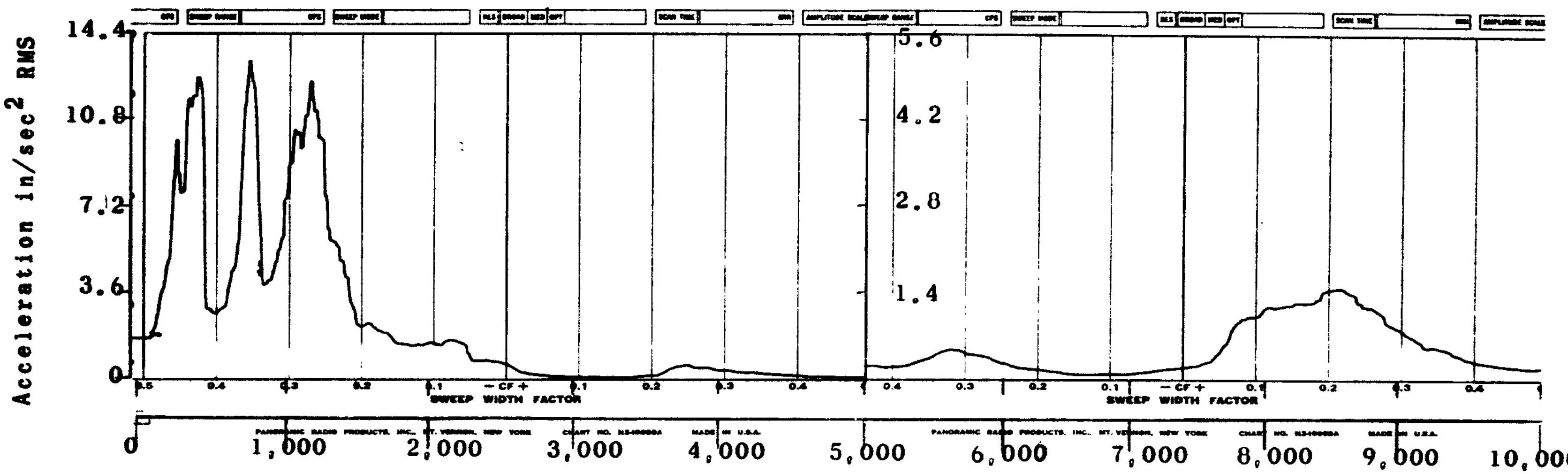


ENCLOSURE 6 ACCELERATION-SPEED EXPONENT FOR THE SEVERAL OCTAVE BANDS AT 10,000 LBS. LOAD FOR THE NJ240-A CYLINDRICAL ROLLER BEARINGS.

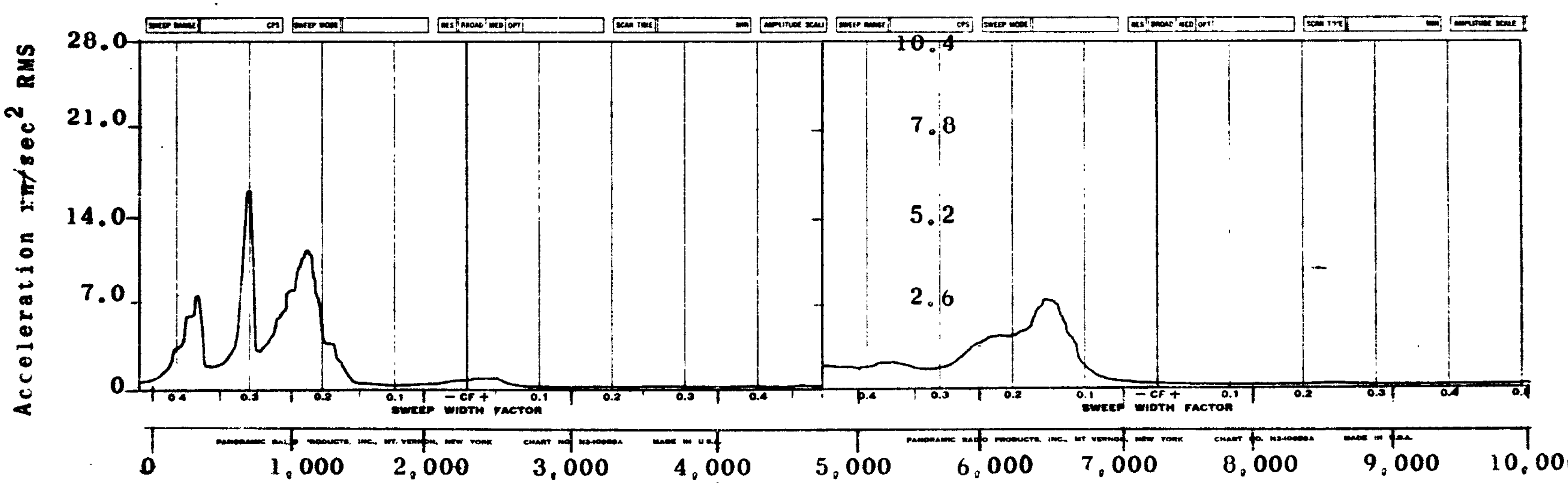


ENCLOSURE 7 OCTAVE BAND VIBRATION SPECTRUM OF NJ240-A AND NJ256-3 CYLINDRICAL ROLLER BEARINGS IN THE VERTICAL DIRECTION AT 300 RPM.

Direction Normal to Load   CRL Accelerometer 504-135



Direction of Load   CRL Accelerometer 504-136



ENCLOSURE 8 VIBRATION SPECTRA OF NJ240-A BEARING UNDER 10,000 LBS RADIAL LOAD AND AT 300 RPM (0-10,000 cps).

ENCLOSURE 9

DIAGRAM OF ORDERS OF WAVINESS CORRESPONDING TO VIBRATION GENERATED AT OCTAVE BAND CUTOFF FREQUENCIES FOR NJ240 CYLINDRICAL ROLLER BEARINGS

	50	100	200	400	800	1600	3200	6400	12800 cps
100 RPM									
W_o	76	152	303	606	1212	2424	4848	9697	19394 wpc
W_i	61	119	238	476	952	1905	3810	7619	15238 wpc
W_b	8	16	32	65	129	258	516	1032	2065 wpc
300 RPM									
W_o	23	45	91	182	363	727	1454	2909	5818 wpc
W_i	18	36	71	143	286	571	1143	2286	4571 wpc
W_b	2.4	5	10	19	39	77	155	309	619 wpc
800 RPM									
W_o	9	17	34	68	137	274	547	1094	2188 wpc
W_i	7	13	27	54	107	215	430	859	1718 wpc
W_b	0.9	1.8	3.6	7.3	15	29	58	116	232 wpc

Where: W_o = Order of Outer Race Waviness
W_i = Order of Inner Race Waviness
W_b = Order of Roller Waviness

ENCLOSURE 10

WAVINESS OF NJ240K CYLINDRICAL ROLLER BEARING COMPONENTS

(NJ240-A AND NJ240-B)

BEARING COMPONENTS	3-6	6-12	12-24	24-48	48-96	15-30	30-60	60-120	120-240	240-480	480-960	960-1920	1920-3840
	WAVINESS VELOCITY (MICROINCHES/SECOND) AT 200 RPM ROTATIONAL SPEED												
INNER RACES													
NJ240-A	825	500	650	2200	6000	900	3500	9500	20000	20500	20500	16000	9000
NJ240-B	800	350	400	1200	1500	800	1400	1600	3000	7000	15000	28000	24000
OUTER RACES													
NJ240-A	1800	450	1000	1850	4500	1350	2100	7500	19500	31000	34000	48500	18500
NJ240-B	1400	1000	500	1800	1000	1000	1600	1200	2200	4000	8000	21000	4000

	4-8	8-16	16-32	32-64	64-128
	WAVINESS VELOCITY (MICROINCHES/SECOND) AT 740 RPM ROTATIONAL SPEED				
ROLLERS AVERAGE OF 19					
NJ240-A	2750	4350	6250	8850	18000
NJ240-B	520	245	390	620	940

ENCLOSURE 11

RATIOS BETWEEN THE WAVINESS VELOCITIES OF THE IMPROVED BEARING AND THE STANDARD PRODUCTION QUALITY BEARING

Bearing Components						Frequency Bands (wpc)							
	3-6	6-12	12-24	24-48	48-96	15-30	30-60	60-120	120-240	240-480	480-960	960-1920	1920-3840
Inner Ring													
B/A	0.92	0.70	0.61	0.55	0.25	0.88	0.40	0.17	0.15	0.34	0.73	1.74	2.7
Outer Ring													
B/A	0.78	2.21	0.50	0.97	0.22	0.74	0.76	0.16	0.11	0.13	0.24	0.43	2.2

	Frequency Bands (wpc)				
	4-8	8-16	16-32	32-64	64-128
Rollers					
B/A	0.19	0.06	0.06	0.07	0.07

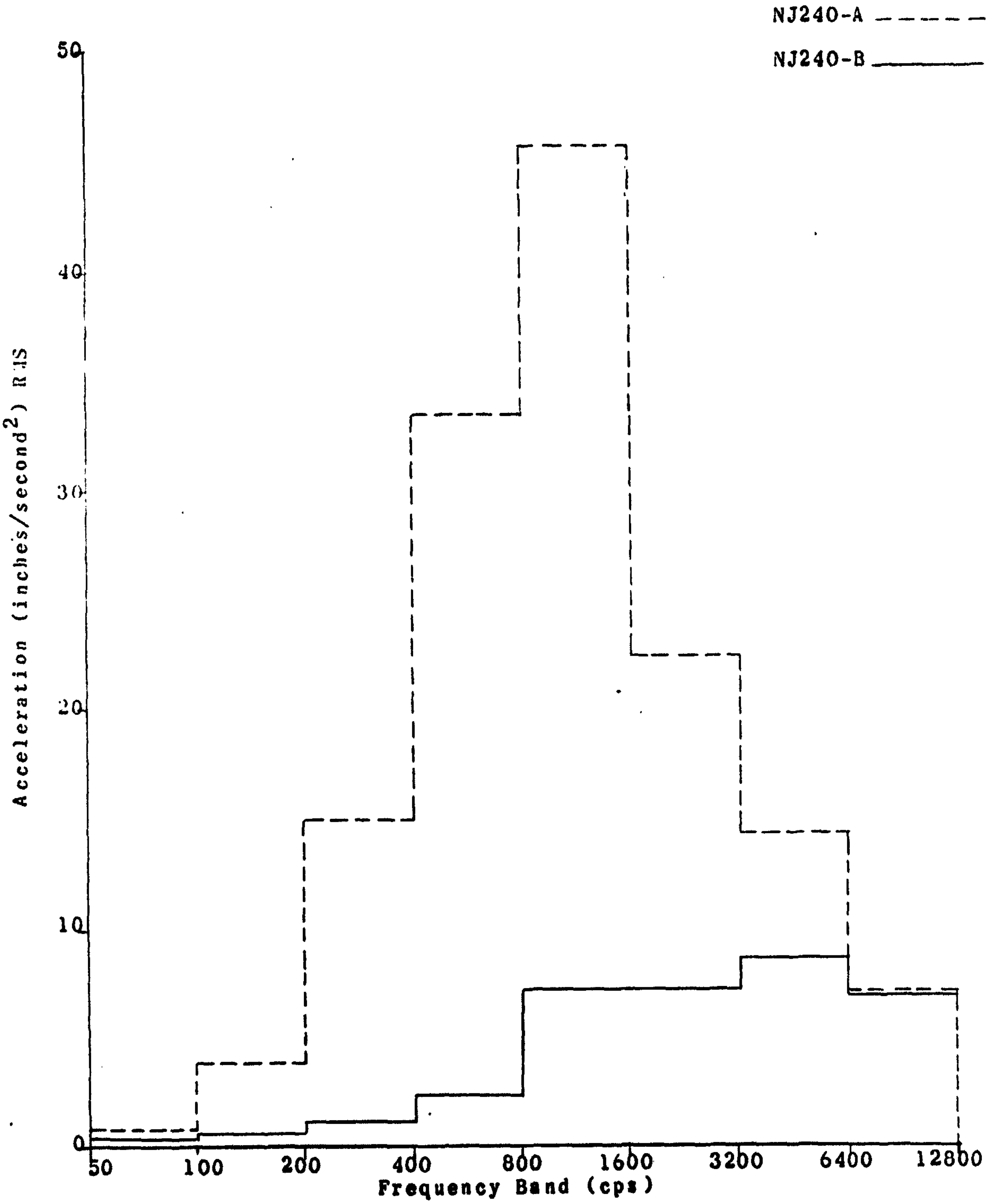
ENCLOSURE 12

VIBRATIONAL ACCELERATION MEASUREMENTS OF THE NJ240 CYLINDRICAL ROLLER BEARINGS AT 10,000 LBS. RADIAL LOAD

Acceleration (inches/second^2) RMS

Frequency Bands (cps)

Speed and Direction		50-100	100-200	200-400	400-800	800-1600	1600-3200	3200 6400	6400 1280
100 RPM									
Horizontal	A	1.97	5.56	8.34	3.48	2.90	2.40	2.88	3.04
	B	0.79	0.76	1.07	0.72	1.74	1.28	2.08	1.22
Vertical	A	1.24	3.97	6.69	2.61	3.10	2.72	4.32	3.04
	B	0.25	0.26	0.56	0.72	1.74	1.47	2.24	2.08
Axial	A	1.09	3.32	14.30	10.50	2.85	2.24	2.40	0.58
	B	0.67	1.23	2.38	1.23	1.52	0.90	0.90	0.29
300 RPM									
Horizontal	A	1.74	7.88	30.20	29.00	38.30	22.40	8.31	8.63
	B	1.11	1.74	2.90	2.67	7.28	7.20	9.60	5.4[illegible]
Vertical	A	0.82	3.72	14.90	33.50	45.80	22.40	14.40	7.8[illegible]
	B	0.37	0.62	1.09	2.36	7.19	7.20	8.64	7.0[illegible]
Axial	A	1.71	3.42	13.30	19.00	22.80	20.80	7.35	1.7[illegible]
	B	2.18	4.46	4.18	2.47	4.75	4.16	4.00	0.9[illegible]
800 RPM									
Horizontal	A	3.02	8.57	19.70	88.10	266.0	208.0	128.0	57.60
	B	2.32	5.68	7.42	6.26	16.20	28.80	99.20	54.40
Vertical	A	3.10	3.96	14.30	91.60	273.0	272.0	244.0	76.80
	B	1.24	1.37	2.73	5.46	17.40	33.60	73.60	48.00
Axial	A	19.90	21.80	19.00	42.70	190.0	154.0	96.00	16.80
	B	20.90	24.70	14.30	6.27	15.20	19.20	28.80	7.68



ENCLOSURE 13 OCTAVE BAND VIBRATION SPECTRUM OF THE STANDARD AND IMPROVED NJ240 BEARINGS IN THE VERTICAL DIRECTION AT 300 RPM.

ENCLOSURE14

RATIO ($\frac{V_B}{V_A}$) BETWEEN THE VIBRATION LEVELS OF THE IMPROVED (NJ240-B) AND THE STANDARD PRODUCTION (NJ240-A) CYLINDRICAL ROLLER BEARINGS

Frequency Bands (cps)

Speed and Direction	50-100	100-200	200-400	400-800	800-1600	1600-3200	3200-6400	6400-12800
100 RPM								
Horizontal	0.40	0.14	0.13	0.21	0.60	0.53	0.72	0.40
Vertical	0.20	0.06	0.08	0.30	0.56	0.54	0.52	0.68
Axial	0.61	0.37	0.17	0.12	0.54	0.40	0.38	0.50
300 RPM								
Horizontal	0.64	0.22	0.10	0.09	0.19	0.32	1.16	0.63
Vertical	0.46	0.17	0.07	0.07	0.16	0.32	0.60	0.90
Axial	1.27	1.31	0.32	0.13	0.21	0.20	0.55	0.57
800 RPM								
Horizontal	0.77	0.66	0.38	0.07	0.06	0.10	0.78	0.95
Vertical	0.40	0.35	0.19	0.06	0.06	0.12	0.33	0.63
Axial	1.05	1.10	0.75	0.15	0.08	0.13	0.30	0.46

ENCLOSURE 15

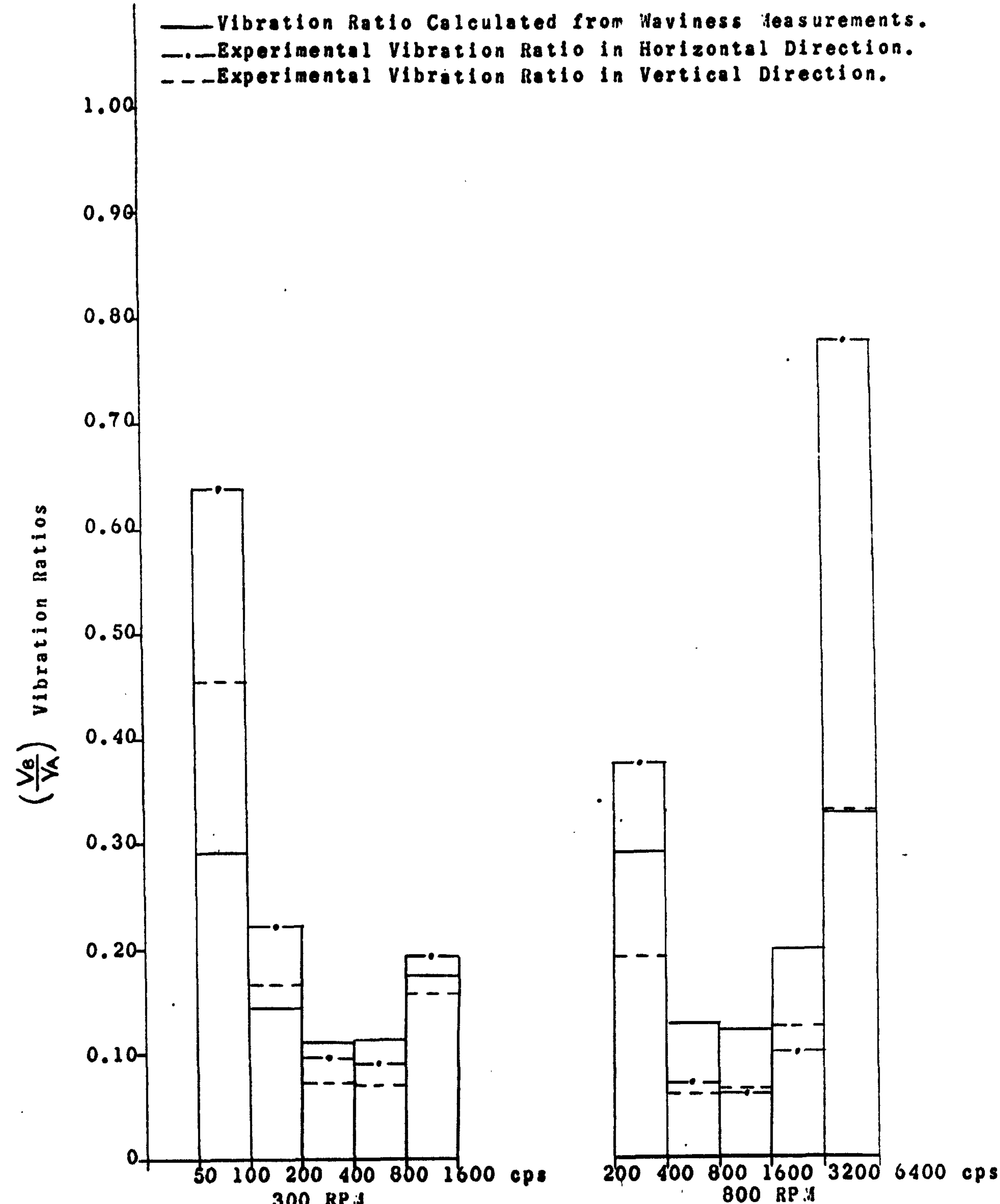
SQUARED AMPLIFICATION FACTORS WHICH SHOW THE RELATIVE INFLUENCE OF OUTER RING, INNER RING AND ROLLER WAVINESS ON THE NJ240-A BEARING VIBRATION LEVEL

Rotational Speed	Frequency Bands (cps) 50-100	100-200	200-400	400-800	800-1600	1600-3200	3200-6400	6400-12800
300 RPM								
Z_o^2	0.04	0.09	0.10	0.21	0.29			
Z_i^2	0.02	0.09	0.25	0.35	0.20			
Z_b^2	0.94	0.83	0.67	0.44	0.51			
800 RPM								
Z_o^2			0.04	0.14	0.27	0.40	0.50	
Z_i^2			0.20	0.37	0.45	0.27	0.14	
Z_b^2			0.76	0.49	0.28	0.33	0.36	

ENCLOSURE 16

PREDICTED FACTORS OF VIBRATION REDUCTION FOR THE TWO NJ240 BEARINGS COMPARED. COMPUTED FROM THE RELATIONSHIP BETWEEN VIBRATION AND WAVINESS GIVEN IN EQUATION 6

	Frequency Bands (cps)							
Rotational Speed	50-100	100-200	200-400	400-800	800-1600	1600-3200	3200-6400	6400-12800
300 RPM								
	0.29	0.14	0.11	0.11	0.17			
800 RPM								
			0.29	0.13	0.12	0.20	0.33	



ENCLOSURE 17 THEORETICAL AND EXPERIMENTAL (Vs/VA) RATIOS FOR THE NJ240 CYLINDRICAL ROLLER BEARINGS IN SEVERAL OCTAVE BANDS AT 300 AND 800 RPM.

ENCLOSURE 18

RATIOS BETWEEN $\frac{V_e}{V_A}$ COMPUTED FROM WAVINESS READINGS AND THE CORRESPONDING VALUES OF $\frac{V_e}{V_A}$ OBTAINED FROM VIBRATION MEASUREMENT (MEAN VALUE OF HORIZONTAL AND VERTICAL VIBRATIONS)

Bearing NJ240-B

	300 RPM	800 RPM
50-100	0.53	
100-200	0.72	
200-400	1.38	1.02
400-800	1.38	2.00
800-1600	0.97	2.00
1600-3200		1.82
3200-6400		0.66

ENCLOSURE 19

SQUARED AMPLIFICATION FACTORS WHICH SHOW THE RELATIVE INFLUENCE OF OUTER RING, INNER RING AND ROLLER WAVINESS ON THE IMPROVED BEARING VIBRATION LEVELS

Speed and Bearing	Frequency Bands (cps) 50-100	100-200	200-400	400-800	800-1600	1600-3200	3200-6400	6400-12800
Bearing B								
300 RPM								
$(Z_o)^2_B$	0.46	0.21	0.20	0.20	0.16			
$(Z_i)^2_B$	0.15	0.67	0.58	0.63	0.75			
$(Z_b)^2_B$	0.39	0.12	0.22	0.17	0.09			
800 RPM								
$(Z_o)^2_B$			0.30	0.23	0.22	0.17	0.27	
$(Z_i)^2_B$			0.37	0.67	0.71	0.79	0.71	
$(Z_b)^2_B$			0.33	0.09	0.08	0.04	0.19	

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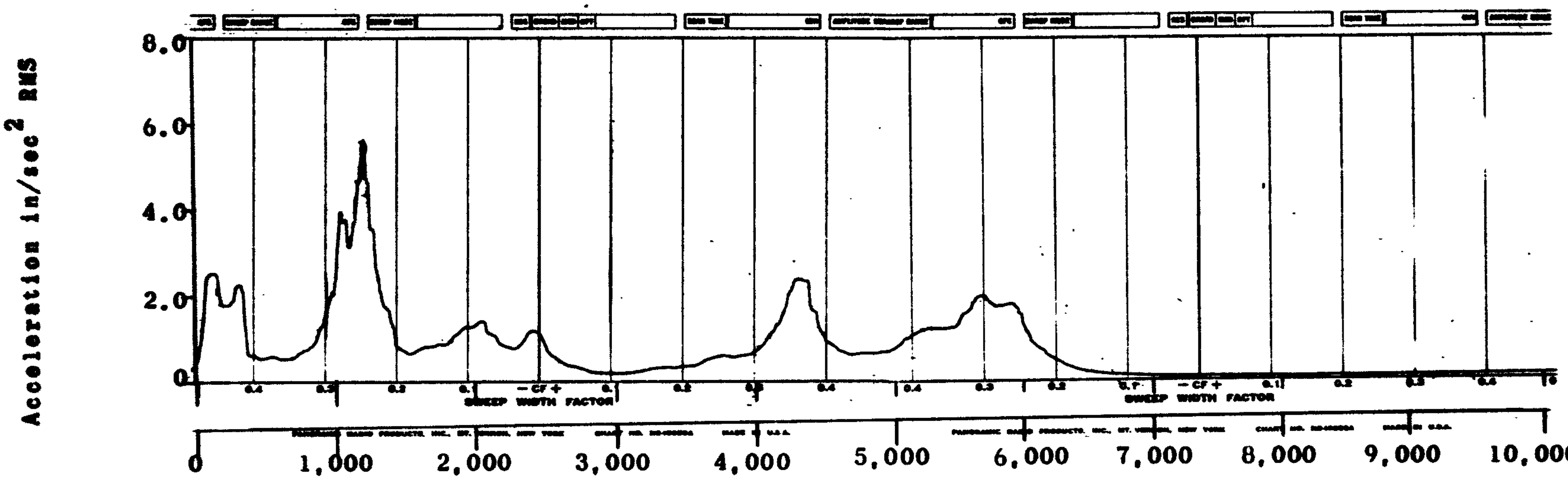
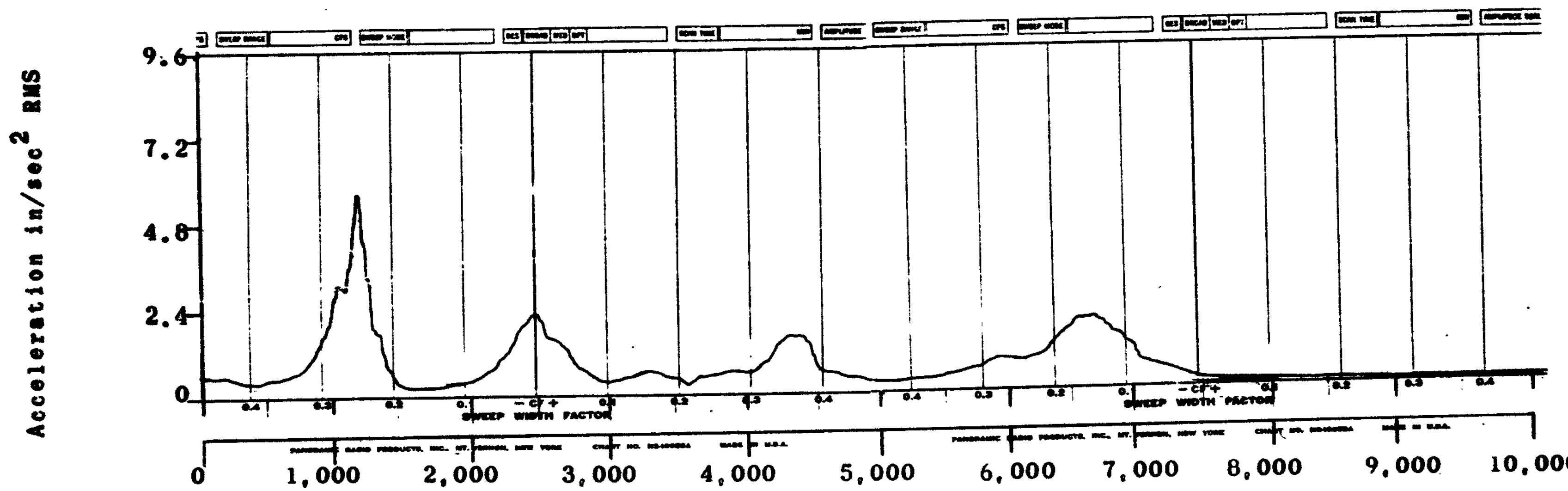
Direction Normal to Load CRL Accelerometer 504-135

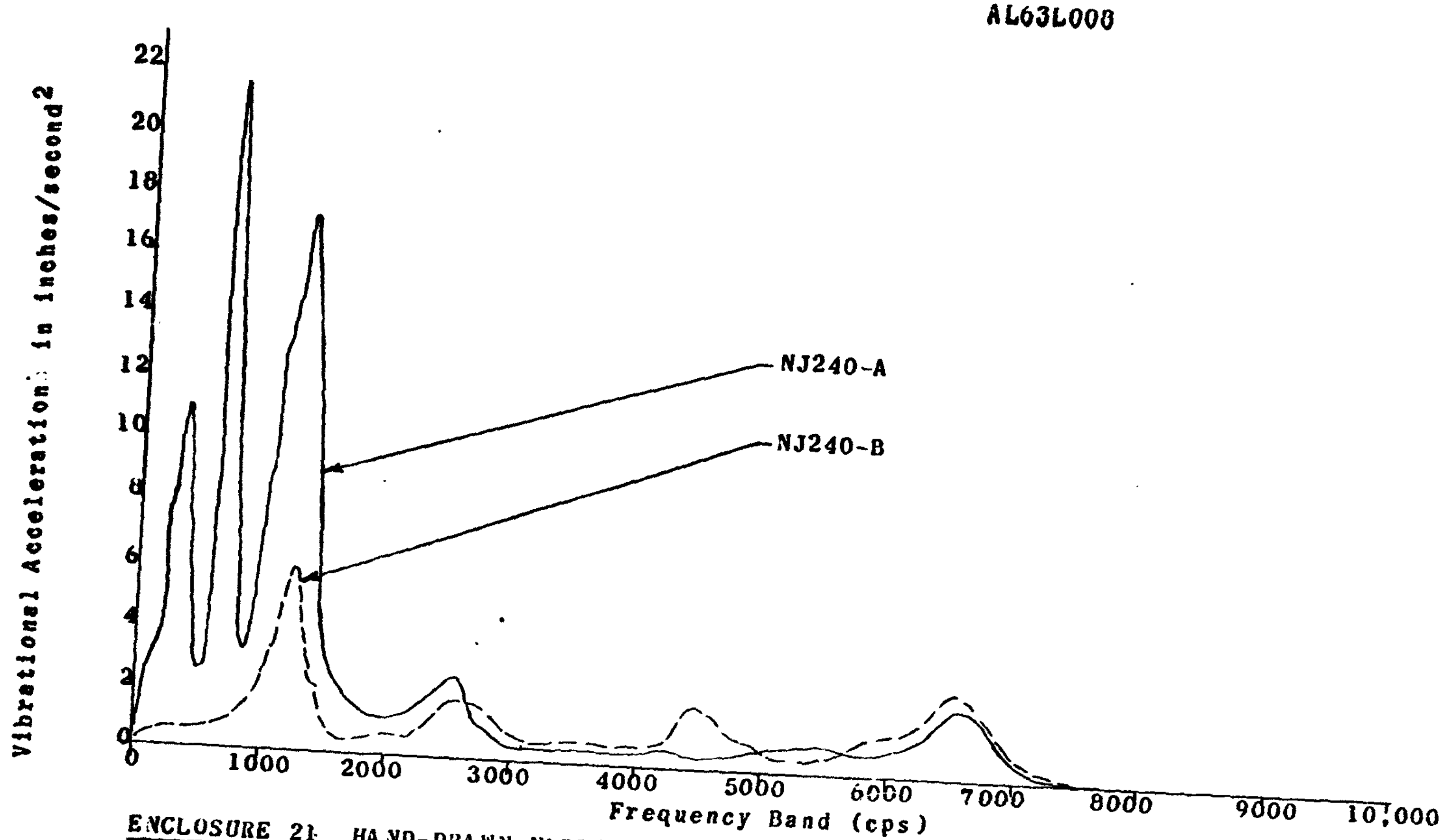


Direction of Load CRL Accelerometer 504-136

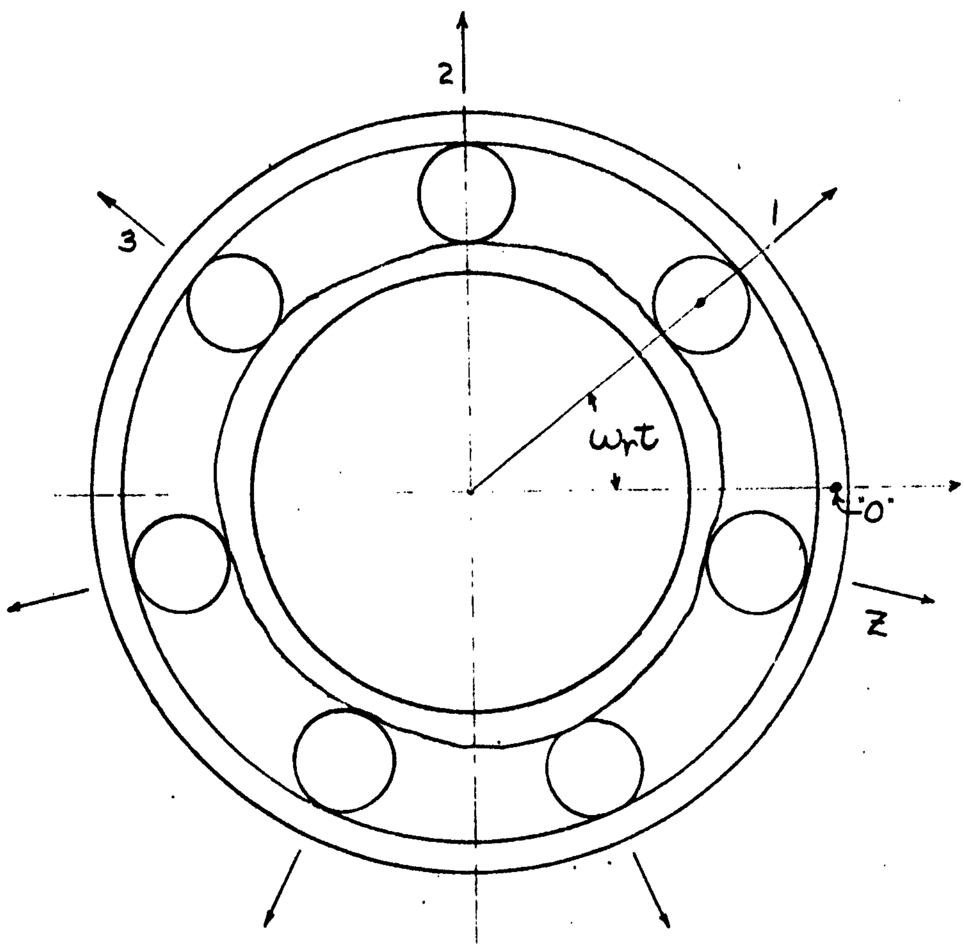


ENCLOSURE 20 VIBRATION SPECTRA OF NJ240-B BEARING UNDER 10,000 LBS RADIAL LOAD AND AT 300 RPM (0-10,000 cps).

AL63L008



ENCLOSURE 21 HAND-DRAWN NARROW BAND SPECTRA OF THE STANDARD AND IMPROVED NJ240 CYLINDRICAL ROLLER BEARINGS IN THE VERTICAL DIRECTION UNDER 10,000 LBS. RADIAL LOAD AT 300 RPM.

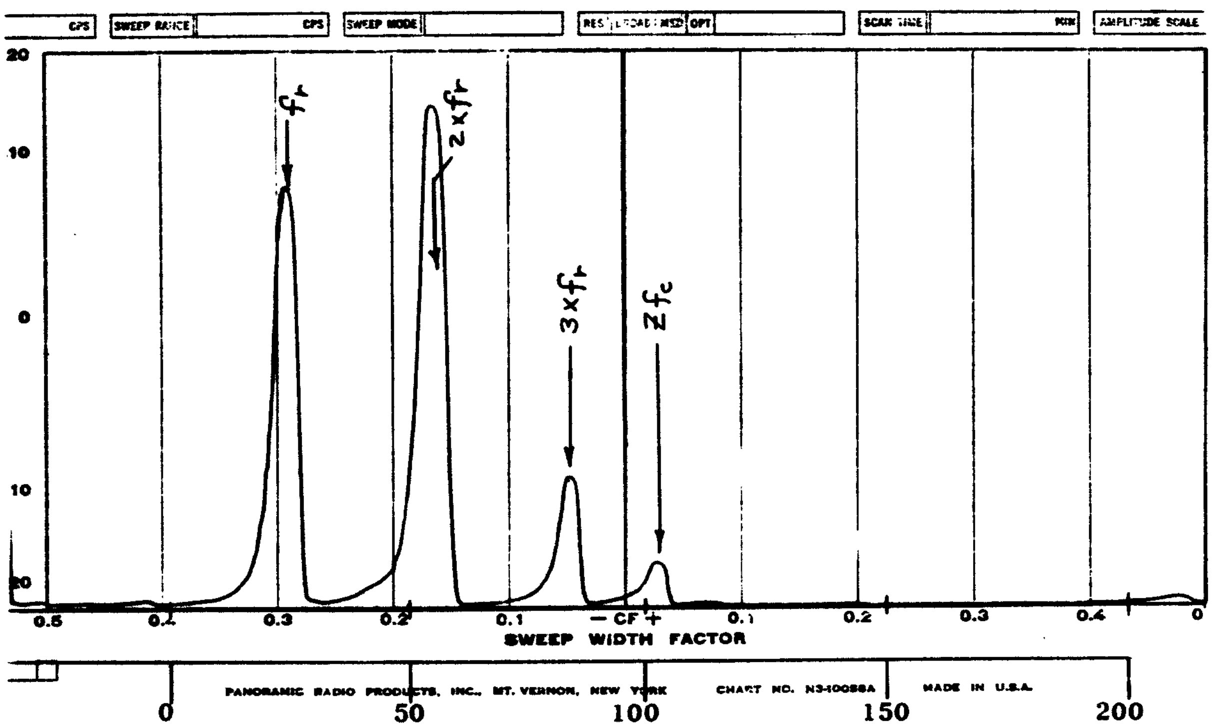


ENCLOSURE 22

AL63L008

2 Point Out of Roundness = 400 Microinches
3 Point Out of Roundness = 10 Microinches

f_r = Rotational Frequency
Zf_c = Ball Pass Frequency

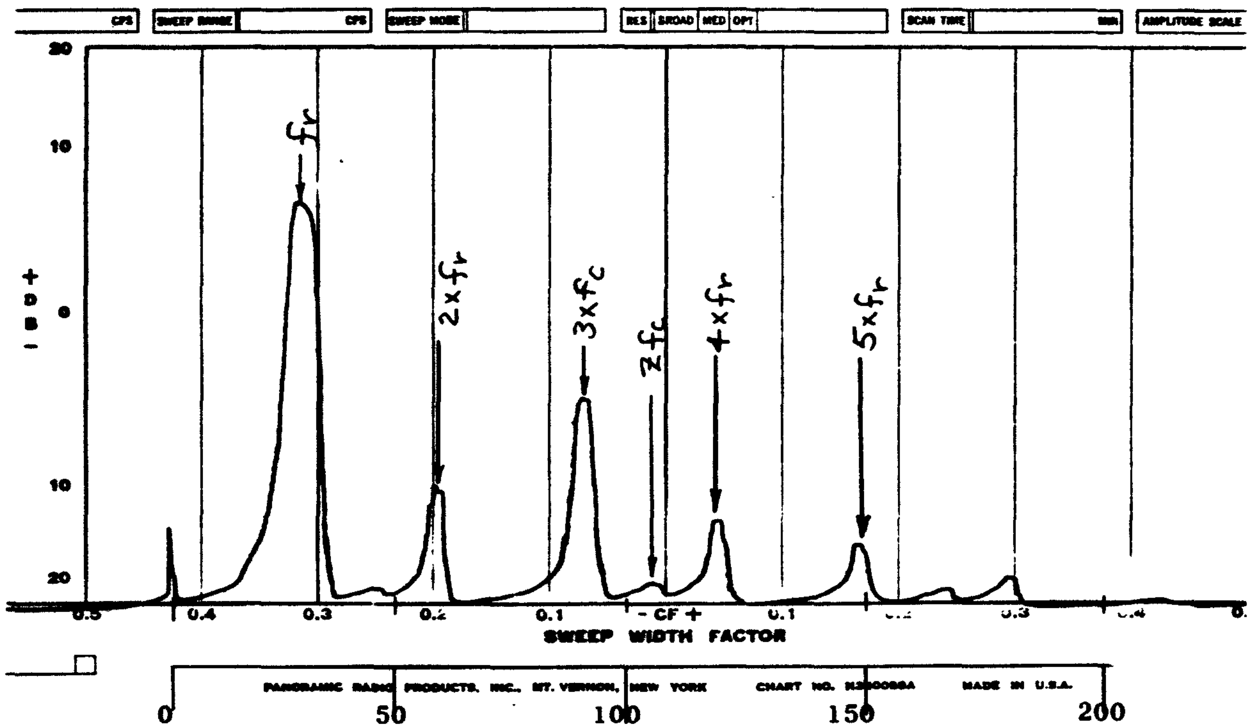


ENCLOSURE 23 NARROW BAND SPECTRA OF A 6205 BALL BEARING AT A ROTATIONAL SPEED OF 1800 RPM IN THE 0-200 CPS FREQUENCY RANGE.

AL63L008

2 Point Out of Roundness = 30 Microinches
3 Point Out of Roundness = 30 Microinches

f_r = Rotational Frequency
Zf_c = Ball Pass Frequency

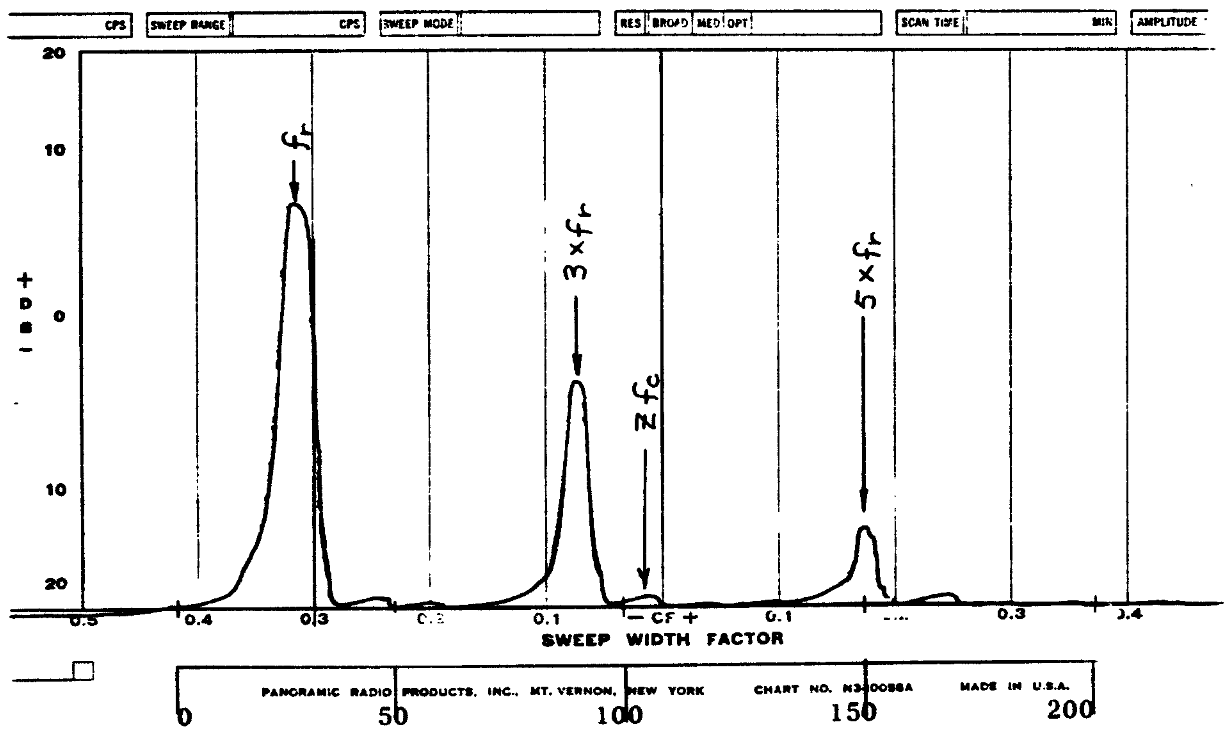


ENCLOSURE 24 NARROW BAND SPECTRA OF A 6207 BALL BEARING AT A ROTATIONAL SPEED OF 1800 RPM IN THE 0-200 CPS FREQUENCY RANGE.

AL63L008

2 Point Out of Roundness = 10 Microinches
3 Point Out of Roundness = 20 Microinches

f_r = Rotational Frequency
Zf_c = Ball Pass Frequency



ENCLOSURE 25 NARROW BAND SPECTRA OF A 6207 BALL BEARING AT A ROTATIONAL SPEED OF 1800 RPM IN THE 0-200 FREQUENCY RANGE.

COMPARISON OF QUIET RUNNING CHARACTERISTICS OF "THICK RING BEARING" AND CONVENTIONAL BEARINGS.

Sample No.	1	2	3	4	5
Type	Thick Ring Bearing	6208	6208	6205	6207
Sample Size	5	50	50	20	20
Ball Size	17/64"	17/64"	5/16"	5/16"	7/16"
Number of Balls	15	8	7	9	9
Flexural Rigidity I/R^3 (in.)	0.00084	0.00021	0.00013	0.00017	0.00015
Bore Diameter (mm)	25	15	15	25	35
Outside Diameter (mm)	62	40	40	52	72
Average Vibration Level in Microinches/second					
50-300 cps band	1340	2745	2463	3153	3140
300-1800 cps band	1700	1598	1517	1774	2465
1800-10,000 cps band	2300	786	500	967	2840
Average Inner Race Waviness in Microinches/second (1000 RPM)					
3-6 WPC	940	2380	765	970	1090
6-12 WPC	630	1156	628	1115	790
12-24 WPC	510	740	598	839	695
24-48 WPC	390	508	756	808	802
48-96 WPC	740	617	1077	1098	1362
Average Outer Race Waviness in Microinches/second (1000 RPM)					
3-6 WPC	2170	3442	1452	1818	1315
6-12 WPC	500	2897	848	2080	1390
12-24 WPC	350	1143	800	1170	908
24-48 WPC	500	678	992	1258	888
48-96 WPC	830	960	1188	1878	2038
Average Ball Waviness in Microinches/second (740 RPM)					
4-8 WPC	55	74	85	108	231
8-16 WPC	72	124	123	185	284
16-32 WPC	99	170	176	188	399
32-64 WPC	147	200	225	288	504
Average Inner Race Two point out of roundness in microinches	32	72	28	32	36
Average Outer Race Two point out of roundness in microinches	88	100	52	44	88
Average Ball Two point out of roundness in microinches	2.0	2.2	2.5	2.8	5.5

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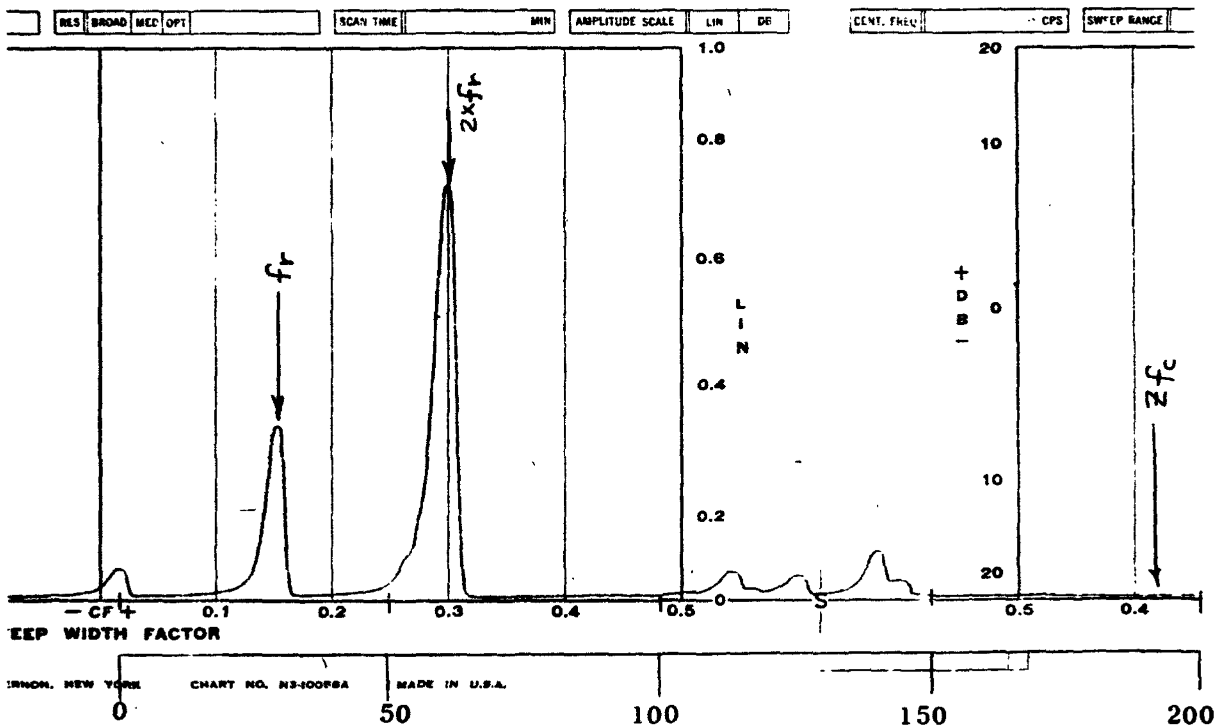
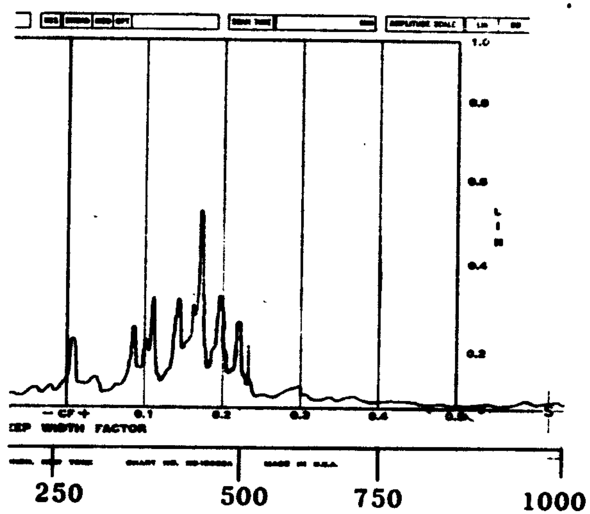
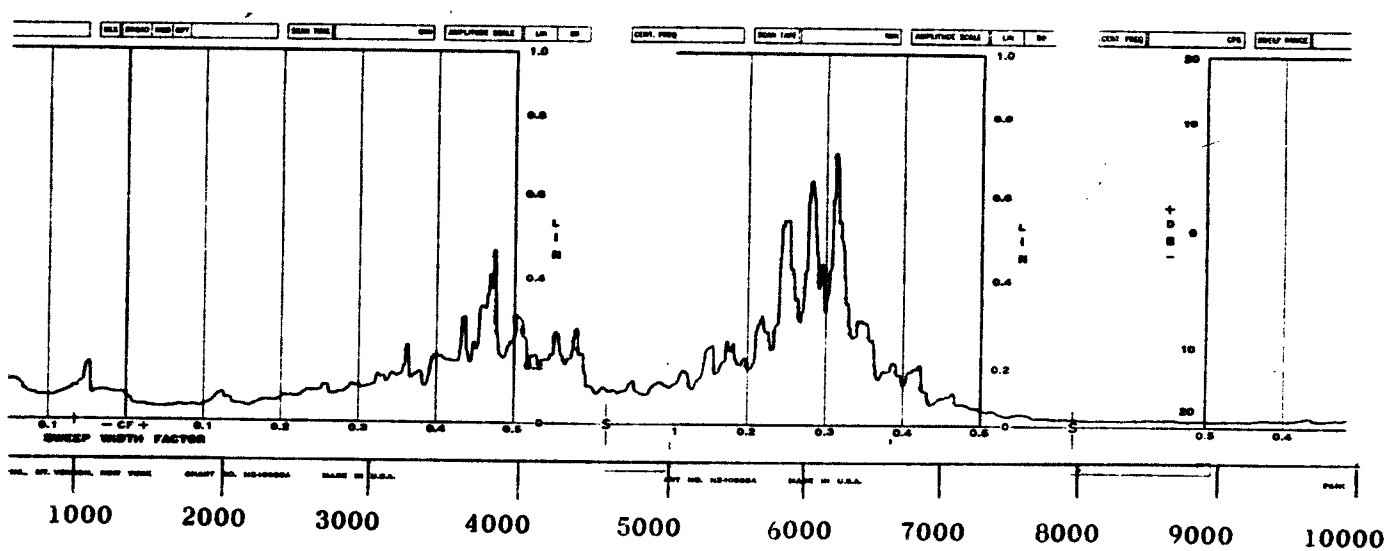
f_r = Rotational Frequency
Zf_c = Ball Pass Frequency



ENCLOSURE 27 NARROW BAND SPECTRA OF THE "THICK RING BEARING" AT A ROTATIONAL SPEED OF 1800 RPM IN THE 0-200 CPS FREQUENCY RANGE





ENCLOSURE 28 NARROW BAND SPECTRA OF THE "THICK RING BEARING" AT A ROTATIONAL SPEED OF 1800 RPM IN THE 200-10,000 CPS FREQUENCY RANGE.